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Demonstration Truck Program 5. Noise Reduction Technology and Costs for a Mack R686 Heavy-Duty Diesel Truck

Bolt Beranek and Newman, Inc.
Cambridge, MA

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Environmental Protection Agency

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Demonstration Truck Program

5

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This is one in a series of seven technical reports and a program summary prepared for the Environmental Protection Agency's Demonstration Truck Program. The reports in this series are listed below.

Report Number	Title	Date
1.	Program Summary: Truck Noise Reduction (BBN Report No. 4839).	December 1981
2.	Noise Reduction Technology and Costs for a Ford CLT 9000 Heavy-Duty Diesel Truck (BBN Report No. 4379).	October 1981
3.	Noise Reduction Technology and Costs for a General Motors Brigadier Heavy-Duty Diesel Truck (BBN Report No. 4507).	October 1981
4.	Noise Reduction Technology and Costs for an International Harvester F-4370 Heavy-Duty Diesel Truck (BBN Report No. 4667).	October 1981
5.	Noise Reduction Technology and Costs for a Mack R686 Heavy-Duty Diesel Truck (BBN Report No. 4795).	December 1981
6.	Field Test of a Quieted Ford CLT 9000 Heavy-Duty Diesel Truck (BBN Report No. 4700).	October 1981
7.	Field Test of a Quieted General Motors Brigadier Heavy-Duty Diesel Truck (BBN Report No. 4796).	December 1981
8.	Field Test of a Quieted International Harvester F-4370 Heavy-Duty Diesel Truck (BBN Report No. 4797).	December 1981

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This report discusses the technology and costs required to reduce the noise of a Mack R686 heavy-duty diesel truck from 81.6 dBA to 73.2 dBA. The noise control treatments consist primarily of a partial enclosure for the engine and transmission, an exhaust silencing system, and two-stage engine mounts. These treatments increase the vehicle weight by 398 lb and estimated vehicle price by $1296.

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NOISE REDUCTION TECHNOLOGY AND COSTS FOR A MACK R686 HEAVY-DUTY DIESEL TRUCK

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PREFACE

This report deals with the technology and costs of treatments developed and implemented by Bolt Beranek and Newman Inc. (BBN) to reduce the noise level of a Mack R686, one of the heavy-duty diesel trucks in the Environmental Protection Agency's Demonstration Truck Program. This program, begun in 1979, included four heavy-duty diesel trucks, each with a different engine. The original program plan called for each vehicle to receive noise reduction treatments and then to enter fleet service for a year of field testing. Each of the four vehicles successfully completed the noise reduction part of the program. The duration of the program was shortened from the original plan; thus only two of the vehicles completed an entire year of field testing. The third truck was in supervised field service for five months, and the Mack R686 did not enter fleet service.

The focus of the Demonstration Truck program was on the technology of treating the vehicles, rather than components such as engines or tires. The EPA conducted parallel programs on diesel engine and tire noise control; these other programs were to be integrated with the truck program. Accordingly, BBN's treatments were primarily to add mufflers for exhaust noise control, enclosures for engine and transmission airborne sound, and vibration isolators for engine structureborne sound where required.

Seven technical reports and a program summary were prepared by BBN for the Demonstration Truck Program. Their titles are listed on the inside cover of this report. The reports appeared in draft version beginning in early 1980 and extending through 1981. The final version of each report was prepared in late 1981. Each of the reports is intended to be internally complete; therefore, some redundancy occurs among the four technology and cost reports. For example, a reader who has already read one technology and cost report will find that he can pass over the nearly identical introduction and test requirements sections (Sec. 1 and Appendix A) and focus on the remaining sections that contain unique technical material.

The authors are grateful to the many governmental and industrial organizations and personnel who have contributed to the development of the noise treatment for this truck. The program has been sponsored by the Environmental Protection Agency's Office of Noise Abatement and Control. Mack Trucks, Inc. provided technical information on the truck. The Donaldson Company supplied major exhaust silencing components, and Tech Weld fabricated many of the engine/transmission enclosure components. Noise testing was done at Hanscom Field with the cooperation of the Charles Stark Draper Laboratories and the Massachusetts Port Authority.

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1. INTRODUCTION

The primary objective of the project described in this report has been to reduce the noise level of a Mack R686ST heavy-duty diesel truck from 81.6 to 72 dBA at 50 ft. This target level, established by EPA, is lower than the level of any heavy diesel truck in current production, and has been reached on only four other roadworthy U.S. trucks in recent history [1-5]. An additional objective, also established by EPA, is to ensure that cab noise levels do not exceed 78 dBA. This level corresponds to proposed interior bus noise level of 80 dBA [6], less 2 dBA to account for manufacturing tolerances.

To be acceptable, the noise treatment must allow the truck to function in a normal manner. Accordingly, the treatments must be durable, interfere as little as possible with maintenance activities, add as little weight as possible, permit continued adequate component cooling, and have minimal impact on engine efficiency. All of these factors may be characterized in terms of equipment and operating costs. Projections of initial equipment costs will be treated here; operating costs will be determined during the course of a subsequent in-service evaluation.

The technical approach to the development of noise treatment for the Mack R686 has involved four major phases:

I. Baseline noise testing
II. Development of noise control treatments
III. Final noise tests
IV. Equipment performance and cost estimation.

In the first phase, the untreated vehicle is noise-tested at EPA's Noise Enforcement Facility at Sandusky, Ohio. The vehicle is then delivered to BBN's facility in Cambridge, Massachusetts, where we conduct exterior noise measurements. Diagnostic tests are also performed to determine contributions from major noise sources (intake, exhaust, tires, engine, and transmission). Quantitative goals for each source are established and compared to the actual contributions. The differences then become the noise reduction objectives that must be achieved by each treatment for the entire vehicle to reach the 72-dBA level.

In the second phase, we develop the noise treatment, which consists primarily of an exhaust silencing system, an engine/ transmission enclosure, and engine vibration isolators. The exhaust system is first laboratory-tested to ensure that it meets our goals and then installed on the truck. An enclosure mockup, built of 1/4-in. Masonite and fiberglass, is tailored to the vehicle. These inexpensive and easy-to-form materials are used because of the cut-and-fit approach that is needed to conform to the complex geometry associated with the truck and its many components.

After a suitable mock-up enclosure is developed and tests are performed to indicate that goals have been met, the enclosure is fabricated from metal and sound-absorptive materials, and installed in a nearly final form. In this phase, some refinements are implemented to tune the system acoustically, thereby bringing the vehicle into closer compliance with the goals.

In Phase III, the truck undergoes final noise testing. Exterior noise levels are measured in accordance with the EPA test procedure [7] and in-cab levels are determined by following the SAE J336a Recommended Practice.

While performance and cost factors are taken into account qualitatively in the numerous decisions made throughout the program, a formal assessment of these factors is deferred until the vehicle is complete. At this point (Phase IV), an analysis of certain performance factors and of equipment costs is performed.

Section 2 presents a description of the baseline configuration and noise levels of the Mack R686. Details of the noise control treatments and their estimated effectiveness are discussed in Sec. 3. Section 4 presents the final interior and exterior noise levels. Estimates of fuel economy impacts, engine mount capacity and serviceability are treated in Sec. 5. Section 6 presents the cost estimates for the treatments. Noise test procedures are briefly summarized in Appendix A. Detailed calculations of the source contributions are presented in Appendices B and C.

2. BASELINE TRUCK CONFIGURATION AND NOISE LEVELS

2.1 Truck Description

The baseline truck, as received by BBN at the beginning of the noise treatment project, is illustrated in Fig. 1. It is a Mack Model R686 regular conventional 6 × 4 tractor with a 151-in. wheel base. The cab is 107 in. long (BBC). Fully fueled, but without a driver, the tractor weighs 15,782 lb; it has a gross combination weight rating (GCWR) of 80,000 lb. Because the truck was built to haul tank trailers, it has provision for

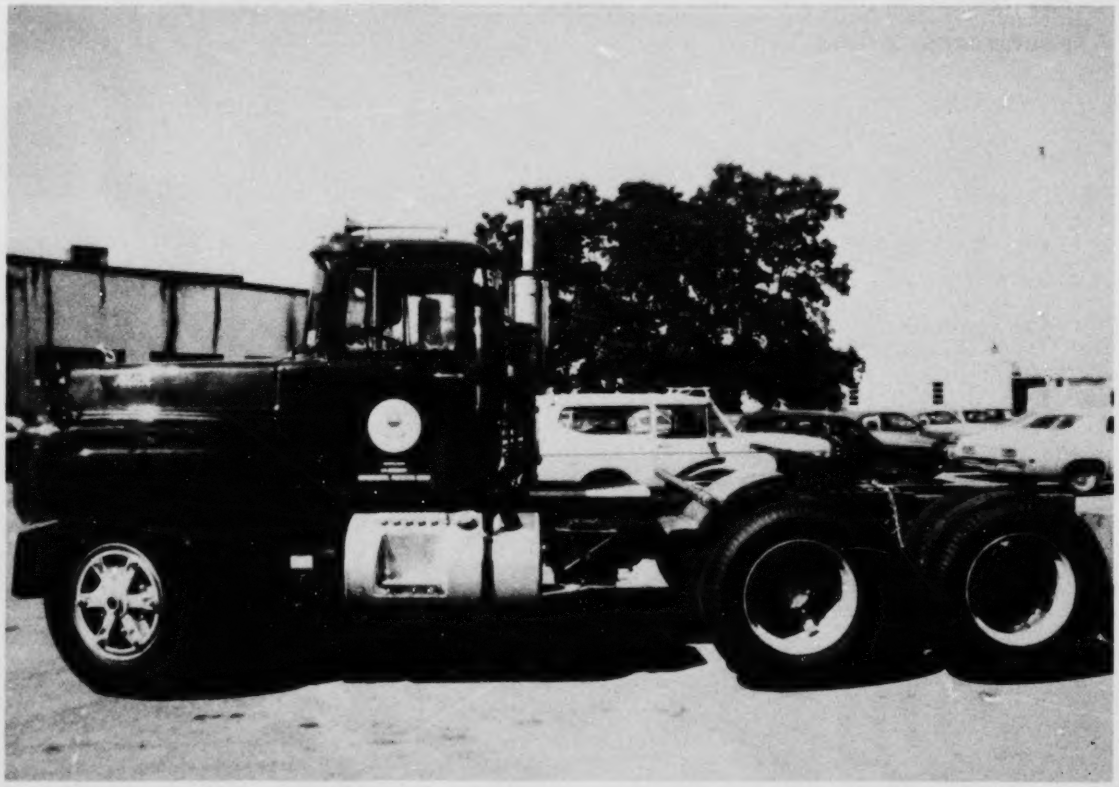

FIG. 1. BASELINE TRUCK CONFIGURATION.

incorporating a turbo unloader* and is equipped with a pump driven from the engine by a power takeoff unit. As we shall discuss in Sec. 3, this equipment significantly impacts the design of the exhaust system and the engine/transmission enclosure.

Figure 1 shows that the baseline truck is equipped with a single vertical exhaust system. The exhaust piping consists of sections of 4-in.-diameter stainless steel flex hose and aluminized steel tubing. The exhaust muffler, Donaldson Model MUM08-5093, has a nominal 8-1/2-in.-diameter unwrapped body and a 34-3/8-in. body length.

Figure 2 presents a closer view of the exhaust system and shows its major components. At the top of the figure may be seen the tailpipe, which is attached to the muffler by means of a U-clamp. The muffler is fastened to the cab by a mounting bracket and to a short flanged pipe by another U-clamp. Probably the most significant feature of the system is the removable section of pipe below the muffler, where a turbo-unloader may be inserted. Provision for the turbo-unloader has clearly restricted the length of the muffler when compared to other stock systems [3,4,5]. The lower sections of piping and the muffler are equipped with heat shields.

The vehicle is equipped with a Mack Model ENDT 676 diesel engine. The engine has a 672 cu-in. (11-L) displacement, is rated at 285 hp at 1800 rpm and is governed at 2100 rpm. It is a 4-stroke-cycle I-6 direct injection engine equipped with a turbo-

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*A turbo-unloader is a turbine-driven pump in which the turbine is powered by the hot exhaust gas and the pump supplies pressurized air to a tank trailer to eject its contents.

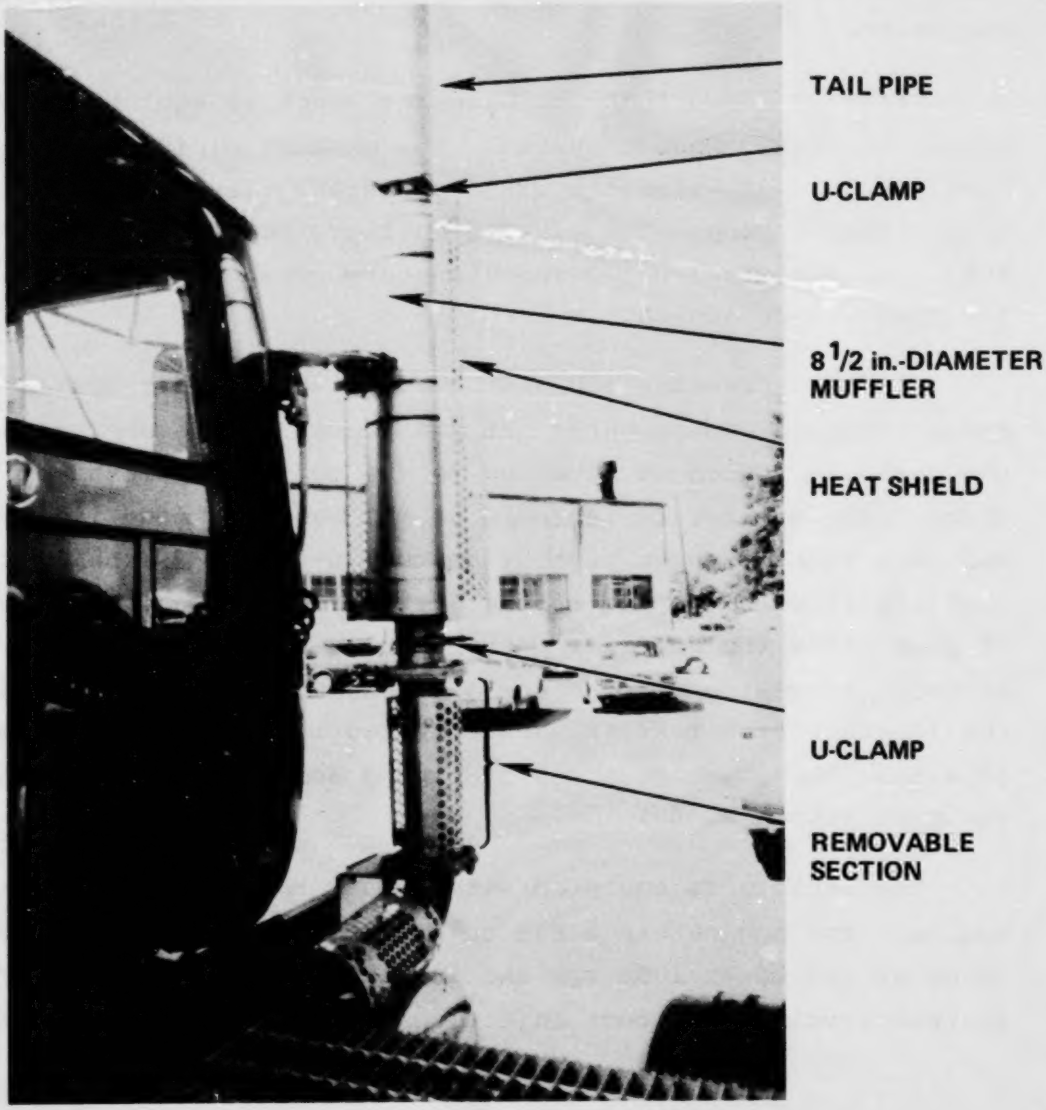
TAIL PIPE
U-CLAMP
8 1/2 in.-DIAMETER MUFFLER
HEAT SHIELD
U-CLAMP
REMOVABLE SECTION

FIG. 2. MAJOR EXHAUST SYSTEM COMPONENTS.

charger and air-to-air intercooler. Figure 3 shows the engine, intercooler outlet, and several other major components of the vehicle.

Engine intake air enters through an externally mounted air cleaner as illustrated in Fig. 4. The bottom duct leads from the cleaner to the turbocharger; the top duct leads to the air-to-air intercooler. The air cleaner is a Donaldson Model EBA15-0048.

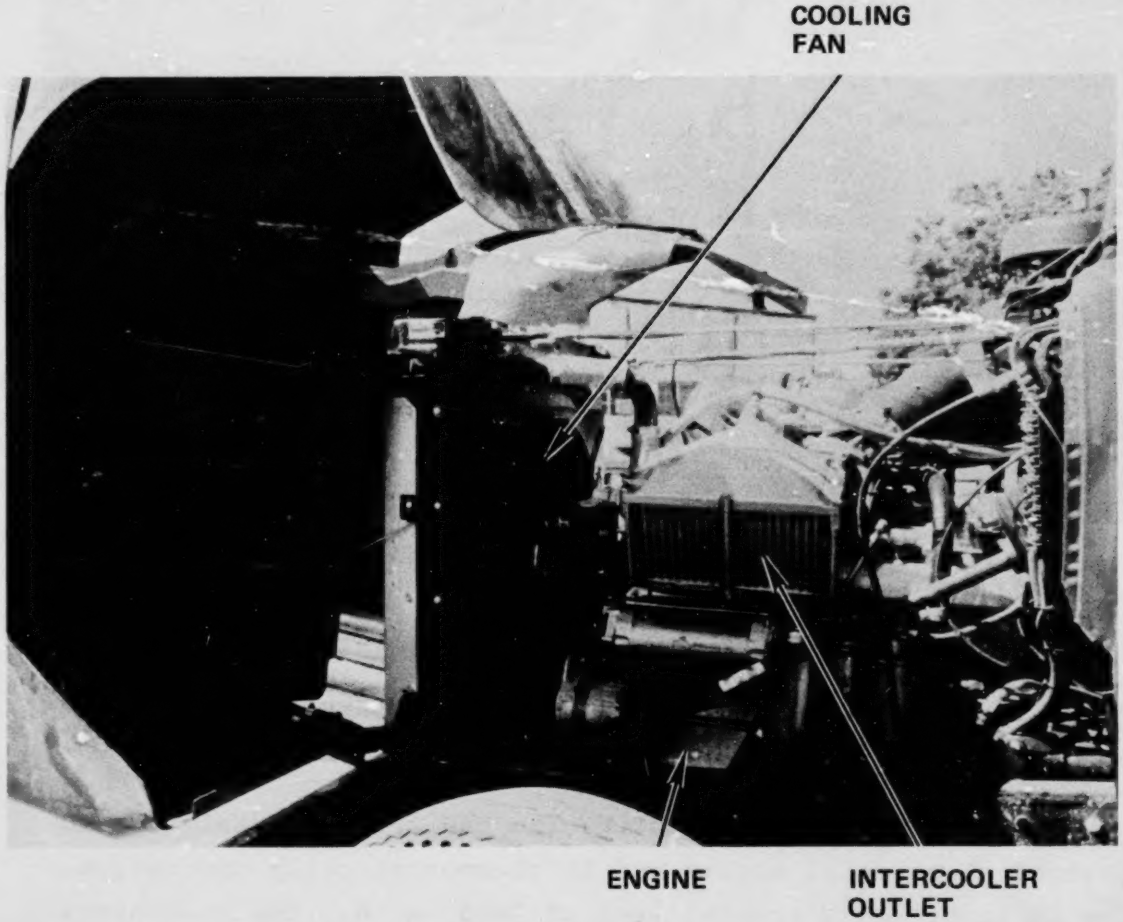

FIG. 3. LEFT SIDE OF TRUCK WITH HOOD TILTED FORWARD TO SHOW SEVERAL MAJOR COMPONENTS.

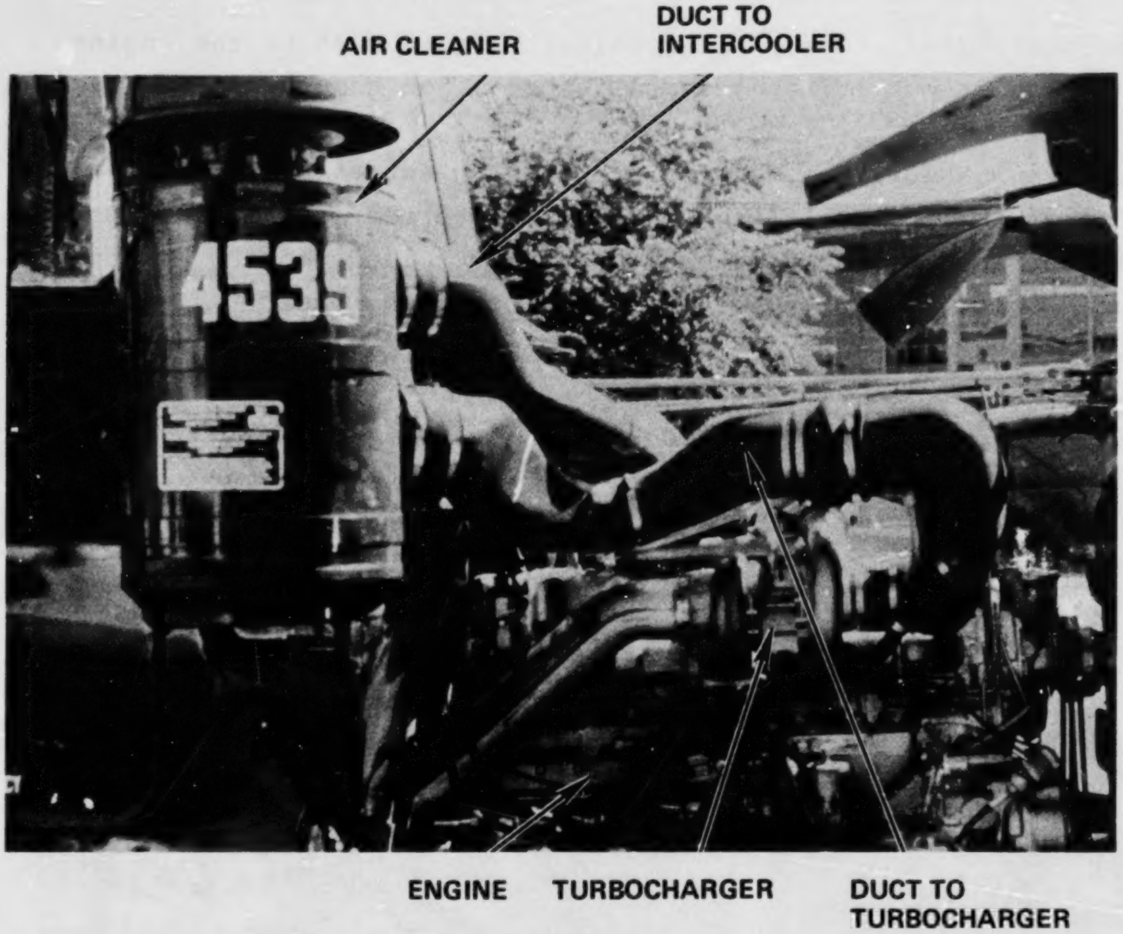
AIR CLEANER
DUCT TO INTERCOOLER
4539
ENGINE
TURBOCHARGER
DUCT TO TURBOCHARGER

FIG. 4. RIGHT SIDE OF TRUCK SHOWING AIR INTAKE SYSTEM AND MAJOR COMPONENTS.

The 24-in.-diameter cooling fan has seven unevenly spaced stamped sheet metal blades and is thermostatically controlled. The radiator has a frontal area of 1000 sq in. The transmission is manufactured by Mack and has 5 forward speeds. The tandem drive rear axles have a 3.73 speed ratio.

All wheels were equipped with 11 × 24.5 radial tires with ribbed tread patterns. These tires were selected for their noise levels, which are lower than those of the cross bar tread commonly used on tractor drive axles.

On the baseline truck there is no apparent treatment of engine noise. Unlike the IH F-4370 [5], for example, the R686ST does not incorporate sound-isolating shields or sound-absorption material applied to the fire wall. Clearly, noise levels meet current standards and there is no need for such treatment. It should be noted, however, that the engine is well shielded from the roadside. Figure 5 shows how the inner fenders on both sides of the vehicle nearly meet the frame rails and shield most of the wheel well area.

One of the single-stage mounts used in the initial truck configuration used to support the engine and transmission is illustrated in Fig. 6. The mount involves a top bracket bolted to the transmission, a bottom bracket bolted to the frame rail, and intermediate rubber isolators. Four isolators (only one of which is visible in Fig. 6) are pressed into the frame rail bracket - two from above and two from beneath. Washers are located above the top isolators and below the bottom isolators to distribute the load exerted by the transmission bracket, and nuts are fastened to the through bolts illustrated in Fig. 6. This design provides rubber isolation for all degrees of freedom while the bolts hold the engine and frame rail mounts securely together.

2.2 Baseline Noise Levels

The truck was initially noise-tested by EPA at its Noise Enforcement Facility at Sandusky, Ohio, and subsequently by BBN at Hanscom Field in Bedford, Massachusetts. Both tests were performed in accordance with the test procedure prescribed by EPA in 40 CFR 205 [7]. This test is very much like the SAE J366b

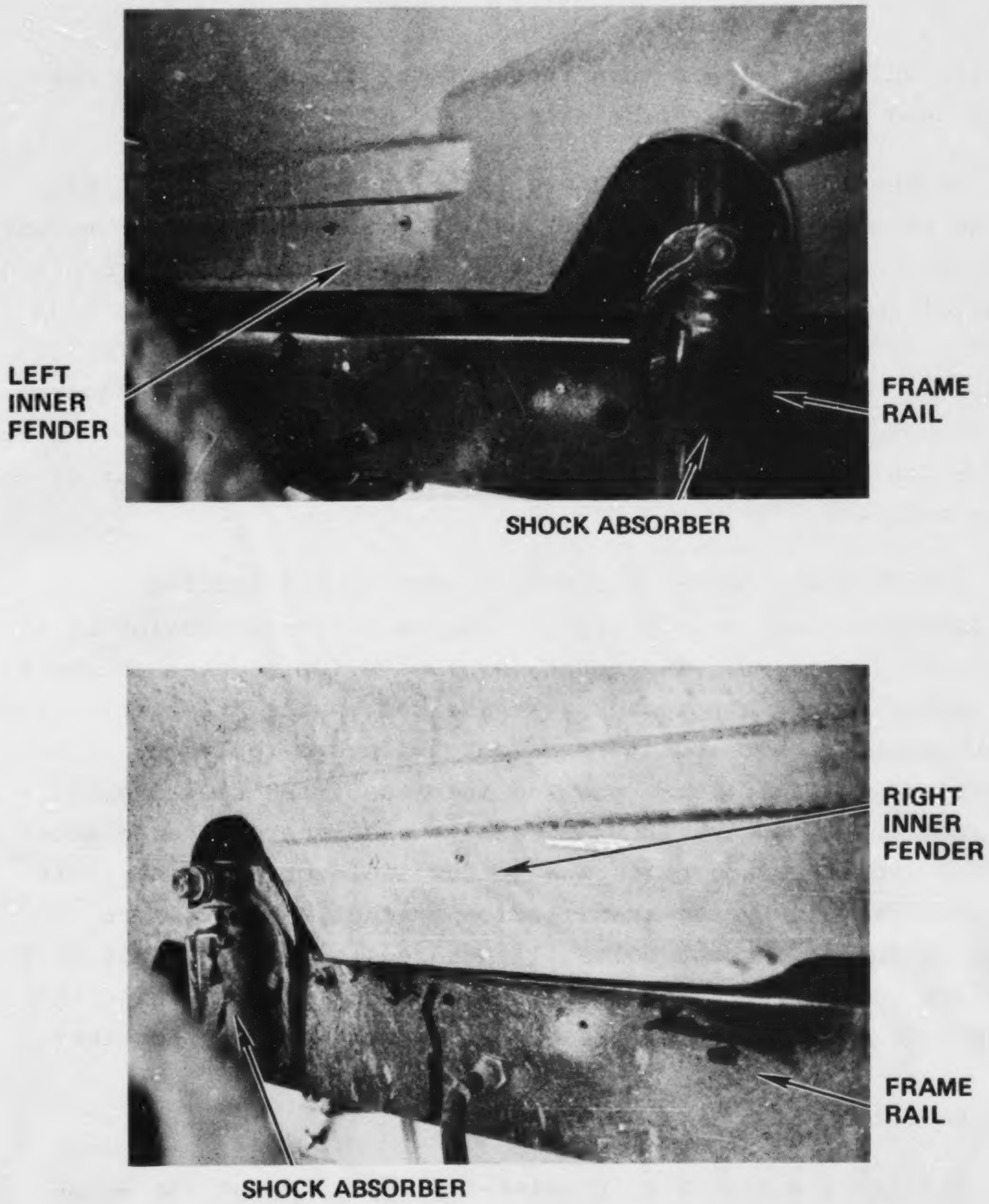
LEFT INNER FENDER
FRAME RAIL
SHOCK ABSORBER
RIGHT INNER FENDER
FRAME RAIL
SHOCK ABSORBER

FIG. 5. FENDERS ON LEFT (TOP PHOTO, TAKEN FROM BEHIND WHEEL) AND RIGHT (BOTTOM PHOTO, TAKEN FROM FRONT OF WHEEL) SIDES OF TRUCK.

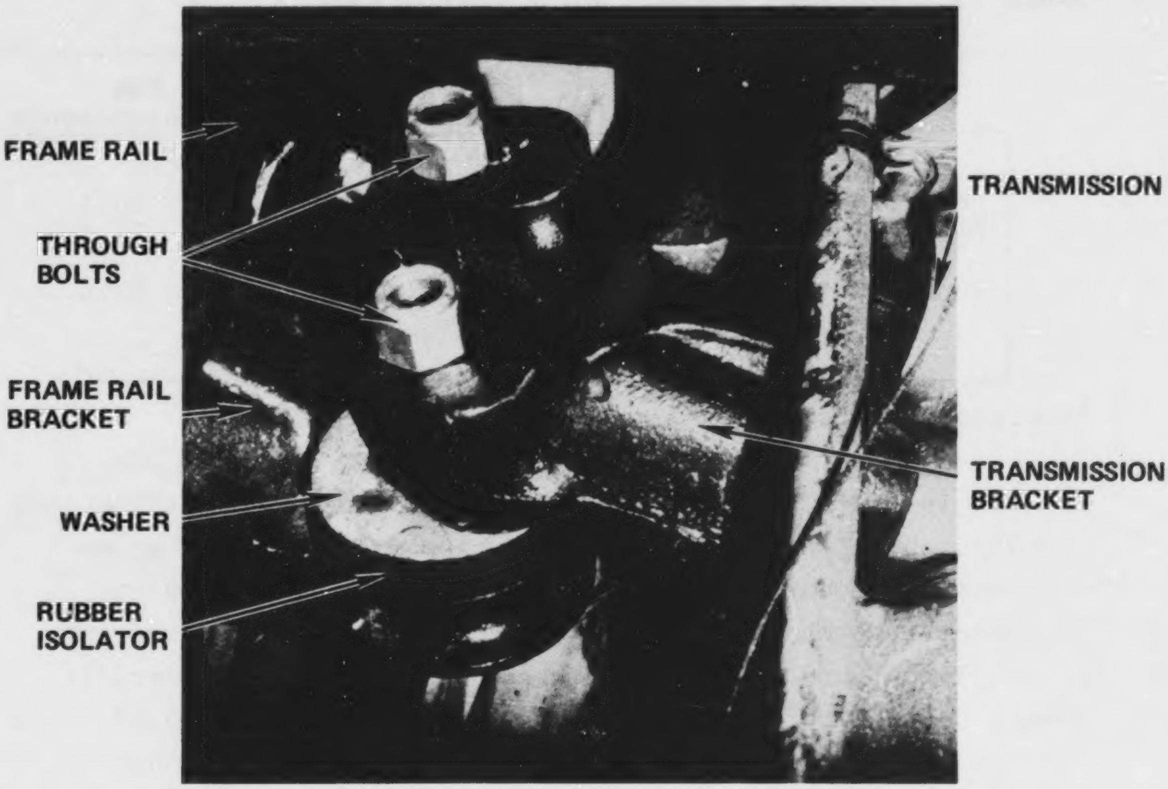
FRAME RAIL
THROUGH BOLTS
FRAME RAIL BRACKET
WASHER
RUBBER ISOLATOR
TRANSMISSION
TRANSMISSION BRACKET

FIG. 6. INSTALLED SINGLE-STAGE ENGINE MOUNT, VIEWED FROM ABOVE. (ONLY ONE OF THE FOUR RUBBER ISOLATORS CAN BE SEEN.)

test; it involves accelerating the vehicle at full throttle from an initial low speed (of about 10 mph for this truck) to a final speed at which maximum governed speed is reached. Noise levels are measured by a microphone located 50 ft from the vehicle's line of travel.

Table 1 shows that the exterior noise levels measured at each location are within one dBA of each other. We will use 81.6 dBA as the baseline level for consistency with most of the tests conducted by BBN.

TABLE 1. BASELINE OVERALL NOISE LEVELS (dBA).

	EPA Measurements	BBN Measurements
Left Side*	81.4	80.1
Right Side*	81.8	81.6
40 CRF 205 Level	81.8	81.6

*Average of first two measurements.

It is useful to know the approximate initial contributions of major noise sources on which to base the design of noise treatments. Laboratory and field tests were conducted to determine the contributions from exhaust, intake, engine and transmission, and tire and aerodynamic sources. However, it should be remembered that while these levels provide guidelines for the development of noise treatments, they are of only secondary importance to the levels of the treated components and complete truck. Therefore, we seek reasonable levels of accuracy (e.g., ±2 dBA) and do not feel that greater precision for these tests would justify significantly greater resource investment than is reported here.

Intake Noise

The baseline intake noise level was measured under laboratory conditions at the Donaldson Company's facility. The experimental configuration is shown in Fig. 7. The laboratory consists of an area inside a building, housing a test engine and dynamometer, and an outdoor area in which key components and a microphone are located. The acoustic wall shown in the figure is part of the building and is constructed of a double wall of

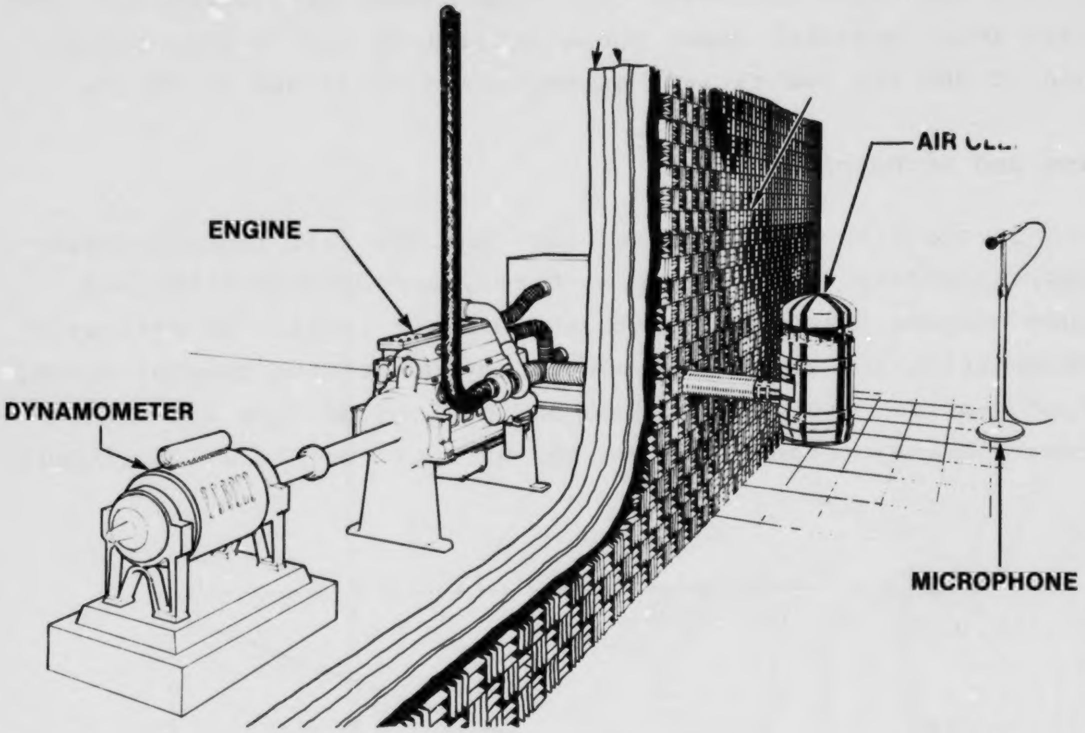

FIG. 7. EXPERIMENTAL CONFIGURATION FOR INTAKE NOISE MEASUREMENT.

concrete and an exterior foam surface. The concrete is sufficiently thick to attenuate noise radiated by the engine to negligibly low levels. The sound-absorbing foam is intended to minimize the contribution of intake noise that is reflected from the concrete wall. The EBA15-0048 air cleaner and air intake duct used in the test are the same models as those installed in the truck.

Because intake noise levels were relatively low, a microphone was placed 75 in. from the intake duct so that an adequate signal-to-noise ratio could be obtained. To simulate the operational conditions that occur during a truck passby test, the engine is accelerated, using only the rotary inertia of the dynamometer as a load. (Donaldson has found that levels measured by this technique correlate well with passby measurements.) The noise level measured under these conditions was 70 dBA, which, when 18 dBA are subtracted, extrapolates to 52 dBA at 50 ft.

Tire and Aerodynamic Noise

In addition to the major noise sources that require treatment, secondary sources such as tires, aerodynamic flow, and other components contribute to the overall level. We estimated the contribution from these sources by conducting coastby tests, which provide particularly good indications of tire and aerodynamic noise. Figure 8 shows the data plotted on a logarithmic

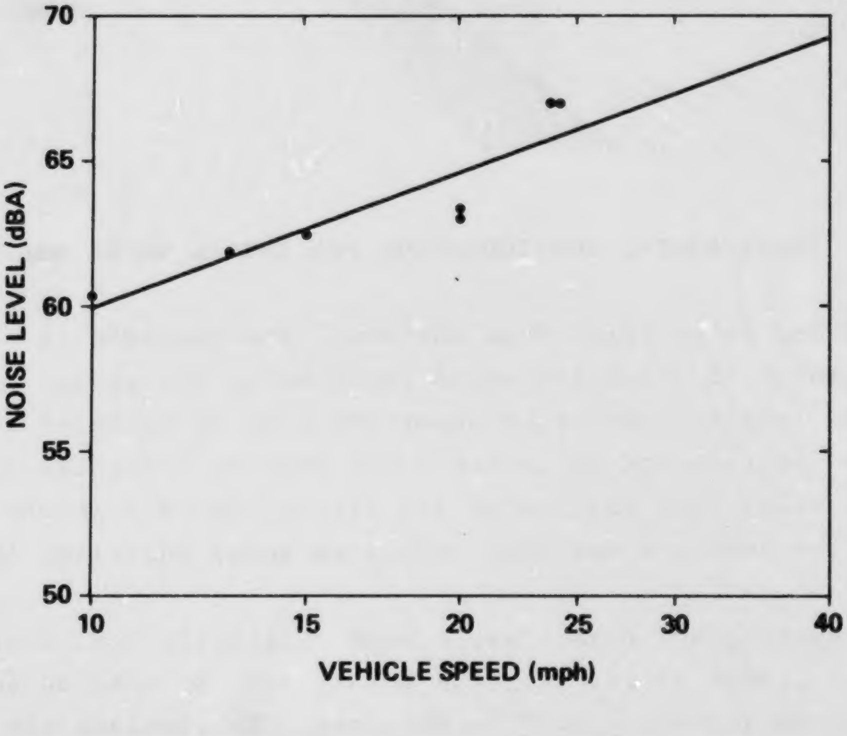

FIG. 8. VEHICLE COASTBY LEVELS.

scale along with a least-squares linear regression curve. The data illustrate that the contribution is approximately 63.5 dBA at the maximum speed of 20 mph reached during 40 CFR 205 tests.

Exhaust Outlet Noise

An estimate of the exhaust outlet noise level was developed from laboratory tests conducted as described above for intake noise measurements. For exhaust noise tests, however, the microphone was located 50 ft from the exhaust stack. The peak level was 70.5 dBA, which occurred during a runup test. As indicated earlier, the results of this type of test correlate well, but not exactly, with vehicle passby test levels.

Exhaust Line Shell Noise

For this vehicle, tests showed that noise radiated from the exhaust line itself was a significant contributor to radiated noise levels when compared with the overall 72-dBA goal. Exhaust line source levels were determined through a diagnostic process in which major components were wrapped with a layer of fiberglass and leaded vinyl and then unwrapped. The results of this test, summarized in Table 2, show that the noise level contributed by the section of line from the turbocharger to the muffler is 61.6 dBA on the left side of the vehicle and 65.6 dBA on the right side. The levels on the left side of the vehicle for wrapped and unwrapped conditions are so close to each other that the difference of 61.6 dBA embodies a high degree of uncertainty. The level difference on the right side is more significant and the resulting estimated pipe contribution is more certain.

TABLE 2. ESTIMATION OF BASELINE EXHAUST SHELL LEVELS.

	Left Side	Right Side
Exhaust pipe unwrapped	72.2	72.1
Exhaust pipe wrapped	71.8	71.0
Estimated pipe level	61.6	65.6

Engine and Transmission Structureborne Noise

Appendix B presents the procedure and results for obtaining an estimate of the noise contributed by truck structural vibration excited by the engine and transmission. The contributed levels are 69.3 dBA for the left side and 66.6 dBA for the right side.

Engine and Transmission Airborne Noise

For this project, the engine and transmission are treated as a single source, around which an acoustical enclosure is to be built. The noise contribution from the engine/transmission combination is estimated by logarithmically subtracting the levels of the other major known sources (exhaust, intake, tires and aerodynamic, truck structure) from the measured overall level of 80.1 dBA for the left side and 81.6 dBA for the right side. The resulting level of 79.0 for the left and 80.9 for the right shows that the engine/transmission levels are very close to the overall level and are the dominant sources of noise.

2.3 Summary of Component Levels

Figure 9 provides an overview of the major noise source levels for the vehicle in its initial, or baseline, configuration

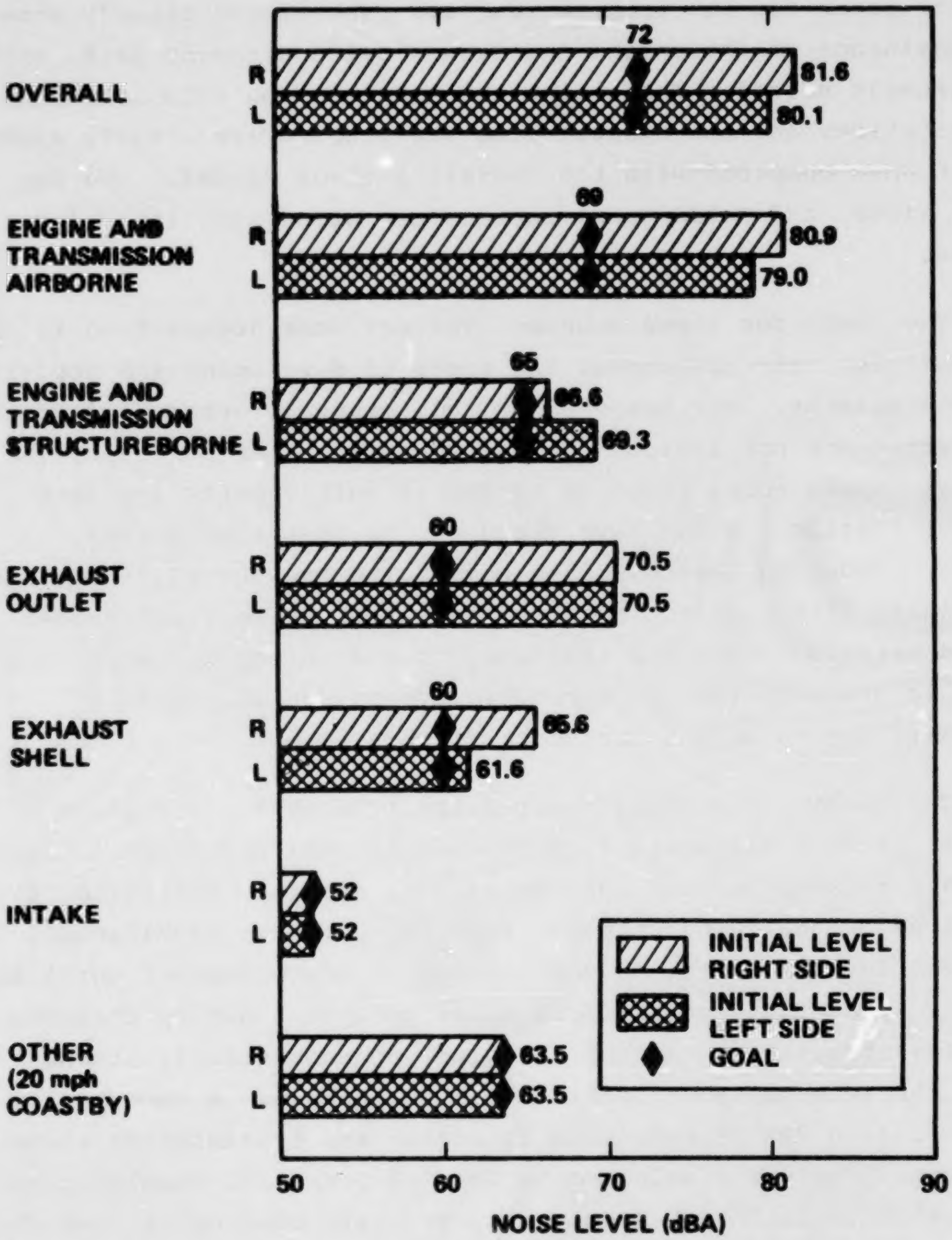

FIG. 9. A-WEIGHTED SOUND LEVELS OF MAJOR SOURCES MEASURED AT 50 FT.

and the goals for the treated sources. The figure clearly shows the dominance of the engine and transmission airborne path, with the exhaust second. The engine and transmission structureborne contributions and the exhaust line shell noise are clearly significant when compared with the overall goal of 72 dBA. The intake, tires, and aerodynamic sources are at substantially lower levels.

The goals for these sources reflect some judgment as to the feasibility, reasonableness, and costs of developing and applying noise treatment. For purposes of this program, intake and coastby sources are not logical candidates for further control. The initial intake noise level of 52 dBA is sufficiently low that further treatment would have virtually no impact on overall levels. Reducing coastby noise beyond the present 63.5-dBA level would have little effect on the total truck noise level associated with the low-speed test used in this program. Moreover, it would probably require tire development, which could be extensive and is beyond the scope of this effort.

The remaining sources all require treatment. The state of the art of flow silencers is sufficiently well developed to make 60 dBA a reasonable goal for the exhaust system. Achieving 10.5 dBA of additional exhaust noise reduction, though significant, is believed feasible with the development of a new exhaust muffling system. Reducing exhaust shell noise by 2 to 6 dBA by changing the pipe structure, wrapping the pipe, or acoustically attenuating the internal sound field seems reasonable. A somewhat modest 2 to 4 dBA of reduction in engine and transmission structureborne levels was selected because of potential complications in further isolating these major power train components from the truck chassis. The overwhelming considerations for structural integrity and alignment generally impose limitations on vibration isolator effectiveness.

All of the above considerations leave a goal of 69 dBA for the engine/transmission combination, which implies a 10- to 12-dBA noise reduction. This reduction may be achievable by means of a partial enclosure but is greater than that required for other vehicles treated in this program.

3. NOISE CONTROL TREATMENTS

The noise control treatments developed for this vehicle encompass (1) an exhaust outlet silencer, (2) exhaust shell treatment, (3) an enclosure for the engine and transmission, and (4) two-stage engine mounts. Here we shall describe and present data on the effectiveness of each treatment.

3.1 Exhaust Outlet Silencing

Unlike the other trucks quieted in this program, the requirement for a turbo-unloader on the R686 effectively precluded the installation of a dual exhaust system. The several options that were considered for locating the turbo-unloader in a dual exhaust system were found impractical. If placed in a branch of the system, the incremental backpressure created by the unloader would undoubtedly force most of the exhaust gas through the other exhaust branch, rendering the unloader ineffective. A shut-off valve for the unused branch is conceptually feasible, but did not appear to be commercially available. Space and accessibility were inadequate to locate an unloader upstream of the flow splitter required by a dual system. Accordingly, we decided to develop a single muffler that, in combination with a stack silencer, would provide adequate noise reduction.

Description

First, a 10-in. cylindrical muffler was developed and tested. The exhaust level was found to have an unacceptably high firing frequency component which required a larger volume than was available for adequate attenuation. To achieve the necessary volume, a muffler with a 10- × 15-in. elliptically shaped cross section was developed. This muffler is shown in a cutaway view in Fig. 10. The exhaust flow enters through the 4-in. exhaust

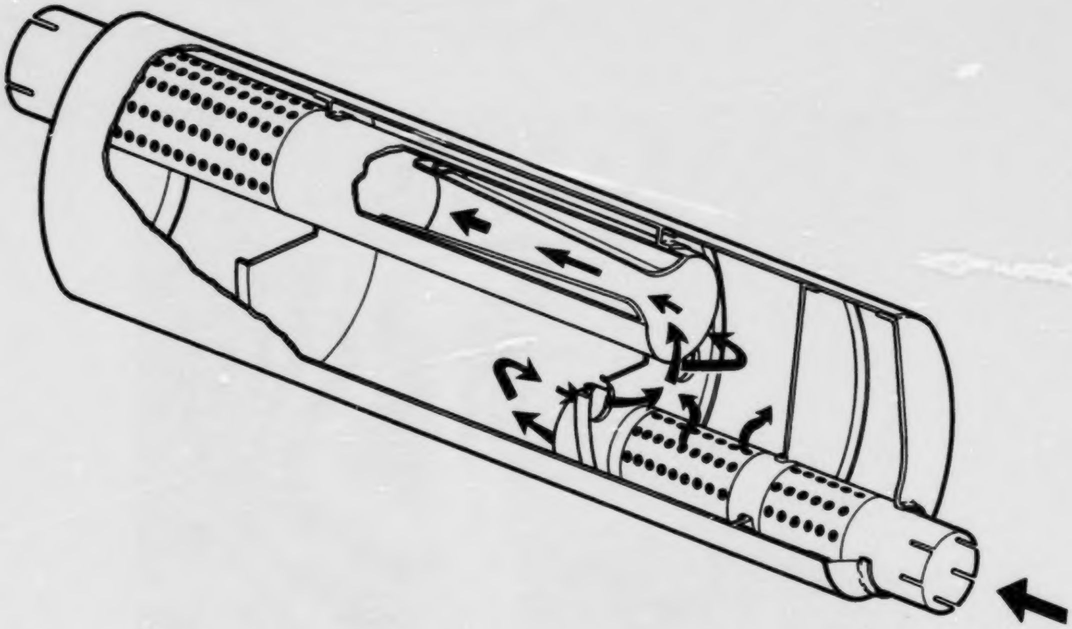

FIG. 10. CUTAWAY VIEW OF 10- × 15-IN. ELLIPTICAL MUFFLER.

pipe at the right of Fig. 10 and passes a resonator made up of a perforated pipe section and an expansion volume. The flow reaches a second resonator and enters a large expansion volume. After being turned twice, the flow passes through a choke, a final resonator, and exits through a 5-in. pipe that is offset from the inlet. Using the 5-in. outlet diameter, rather than the 4 in. used throughout the rest of the system, reduces the exit velocity of the exhaust gas and the concomitant flow noise.

Figure 11 shows the elliptical muffler mounted on the truck. It is placed so that the major axis and the offset exit portion are forward of the inlet. Through this arrangement the muffler does not interfere with trailer clearance requirements, nor does it protrude laterally any more than a standard 10-in. muffler.

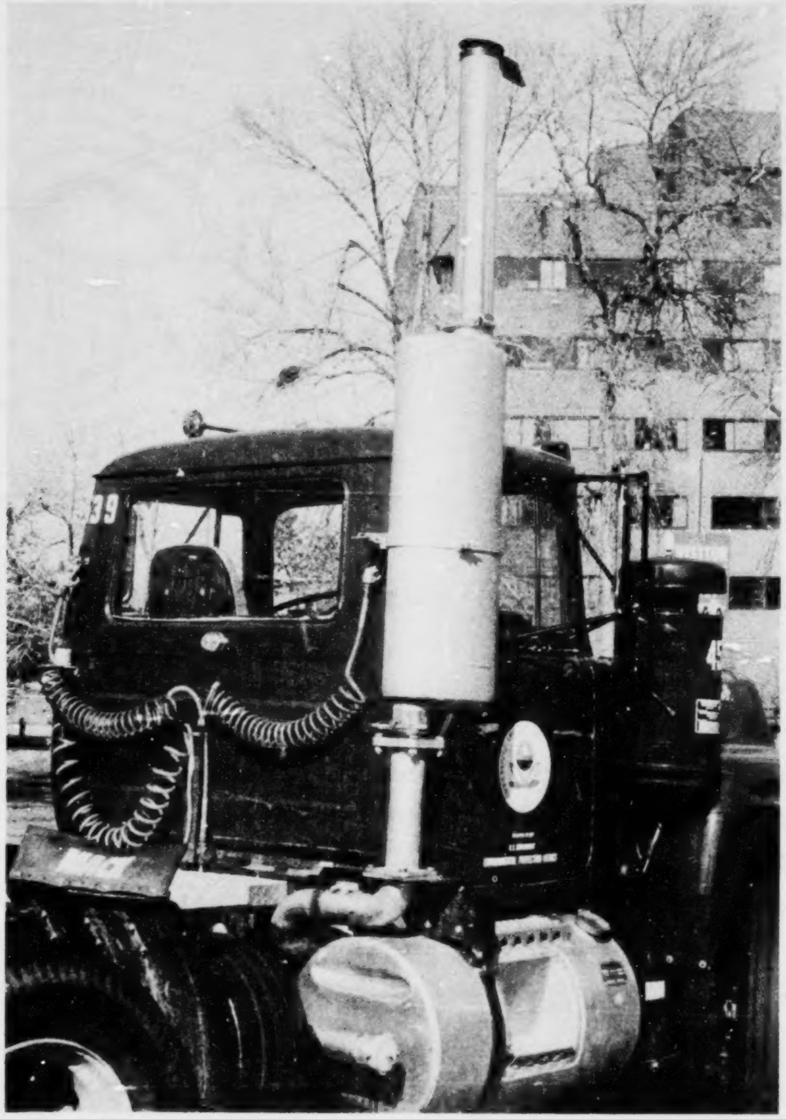

FIG. 11. VIEW OF MUFFLER MOUNTED ON TRUCK.

Also visible in Fig. 11 is a 5-in.-diameter stack silencer just above the elliptical muffler. The stack silencer (Donaldson Model AEM00-1327) has a 4-in.-diameter perforated liner made of aluminized steel, fiberglass packing, and a pressure recovery cone at the outlet.

An additional 5-in. stack silencer was installed in the line between the muffler and turbocharger in order to reduce exhaust pipe shell noise (see Sec. 3.2). This silencer also reduced exhaust outlet noise beyond that achieved by the elliptical muffler and stack silencer alone.

Noise Levels

The exhaust noise level is substantially below the overall truck noise level and cannot be measured readily during a passby test. Accordingly, an indirect measurement must be made and the results used to estimate the passby contribution. We have used two such measurements. One is based on laboratory tests and the other on truck measurements with a microphone located close to the exhaust line outlet. Here we shall discuss each type of test as applied to several exhaust system configurations and then compare test results.

The laboratory tests were conducted with only the elliptical muffler and stack silencer. (It was not until well after these tests were completed that we developed an in-line silencer for shell noise control.) These components were located outside of the same dynamometer test facility used for intake noise measurements as described in Sec. 2. The engine was run up at full throttle to governed rpm and the A-weighted level recorded as a function of engine speed. Because the sound levels were low compared with ambient levels, a microphone was located 18 in. from the centerline of the exhaust stack outlet. The measured level and the estimated level at 50 ft are illustrated in Fig. 12.* A peak level of 56 dBA at 50 ft is reached at approximately 1700 rpm.

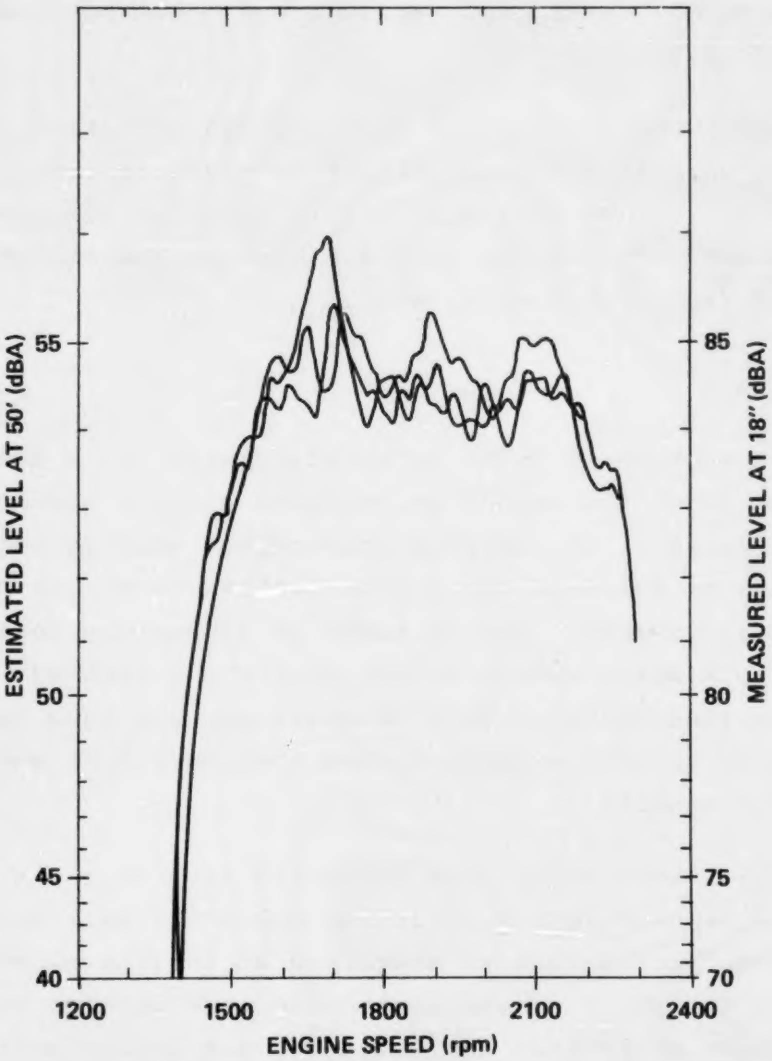

FIG. 12. NOISE LEVELS FOR THE EXHAUST SYSTEM MEASURED DURING THREE RUNUP TESTS.

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*The 50-ft level is estimated by subtracting 20 log (50/1.5) = 30 dB from the level measured at 18 in.

An estimate of the spectrum of the runup sound level was also made. In this case, a runup was performed for each standard octave band from 63 to 8000 Hz and the peak level read from a sound level meter with an integral octave band filter. Each reading is plotted as the A-weighted octave band level shown in Fig. 13.

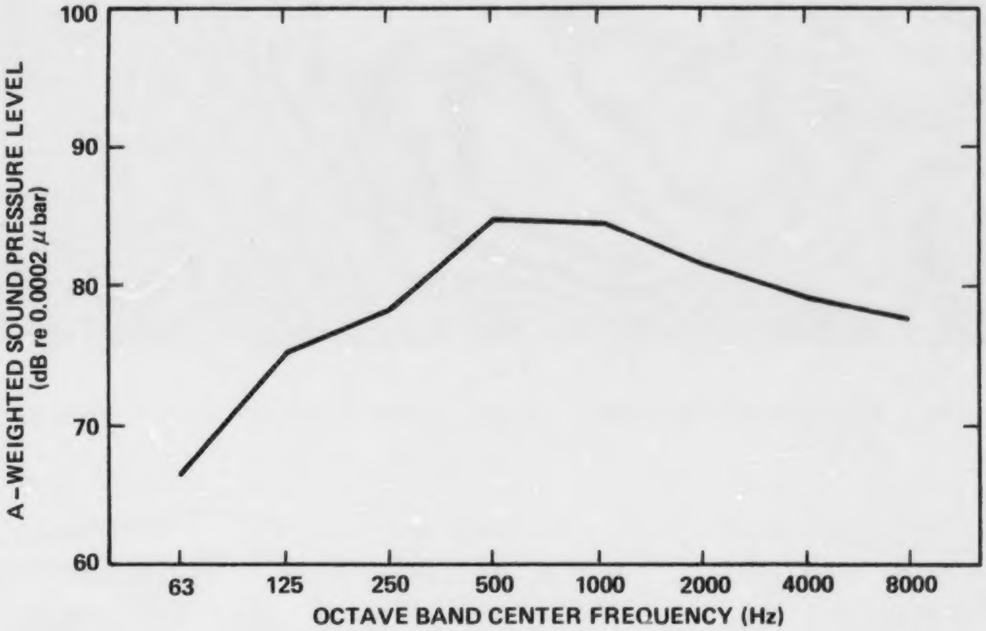

FIG. 13. PEAK OCTAVE BAND LEVELS MEASURED DURING LABORATORY RUNUP TESTS.

Exhaust noise levels were measured on the truck with a microphone located 18 in. outboard of the centerline of the exhaust system (see Fig. 14). One-third octave band spectra for the final configuration (containing an in-line silencer) and an intermediate configuration (without the in-line silencer) are shown in Fig. 15. The in-line silencer appears to provide a modicum of additional low-frequency attenuation and up to 10 dB of additional high-frequency attenuation.

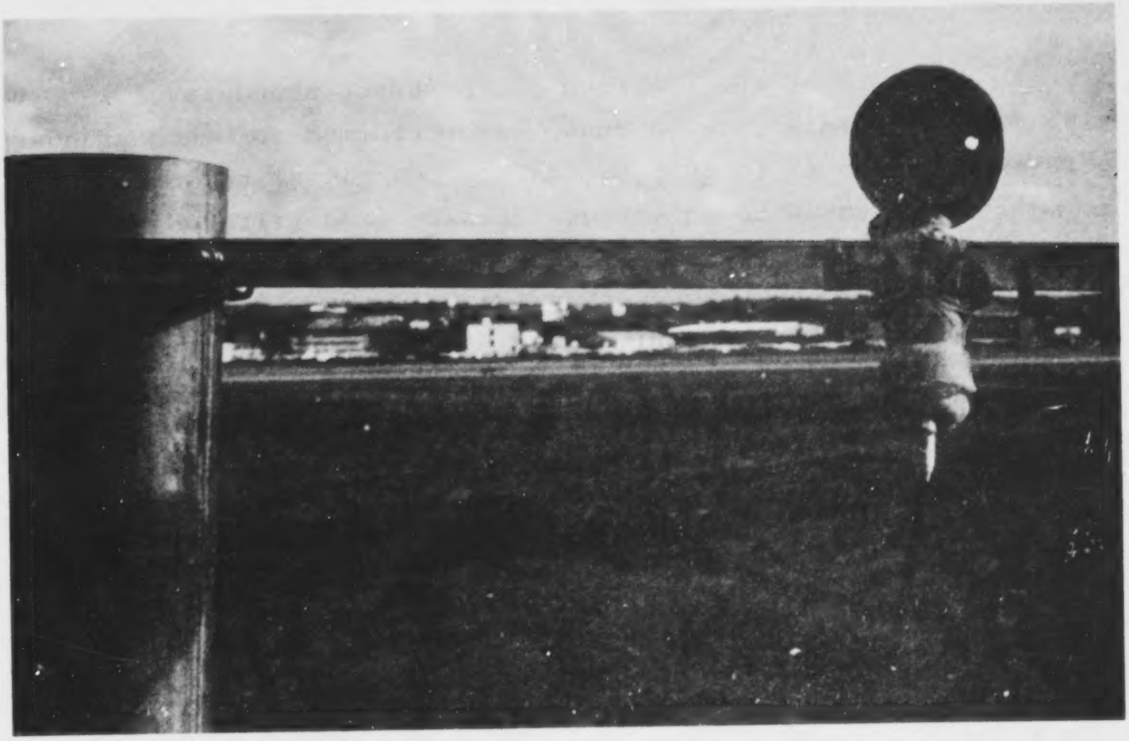

FIG. 14. MICROPHONE LOCATED 18 IN. FROM EXHAUST STACK FOR PASSBY TESTS.

Extrapolating the levels measured at 18 in. to the 50-ft microphone location is done empirically, because of ground reflections and the fact that the propagation path changes constantly during the test. An empirical relation between the level measured at 18 in. and the level measured at 50 ft was found in a separate test. A straight stack was installed on the vehicle to obtain an exhaust-dominated level at both 18-in. and 50-ft microphone locations. The difference between the one-third octave band spectra for both signals gives the transfer function relating the sound at the far microphone to the sound at the near microphone. This transfer function is given in Fig. 16.

From the transfer function in Fig. 16 and the spectra of the sound measured at the microphone 18 in. from the exhaust outlet

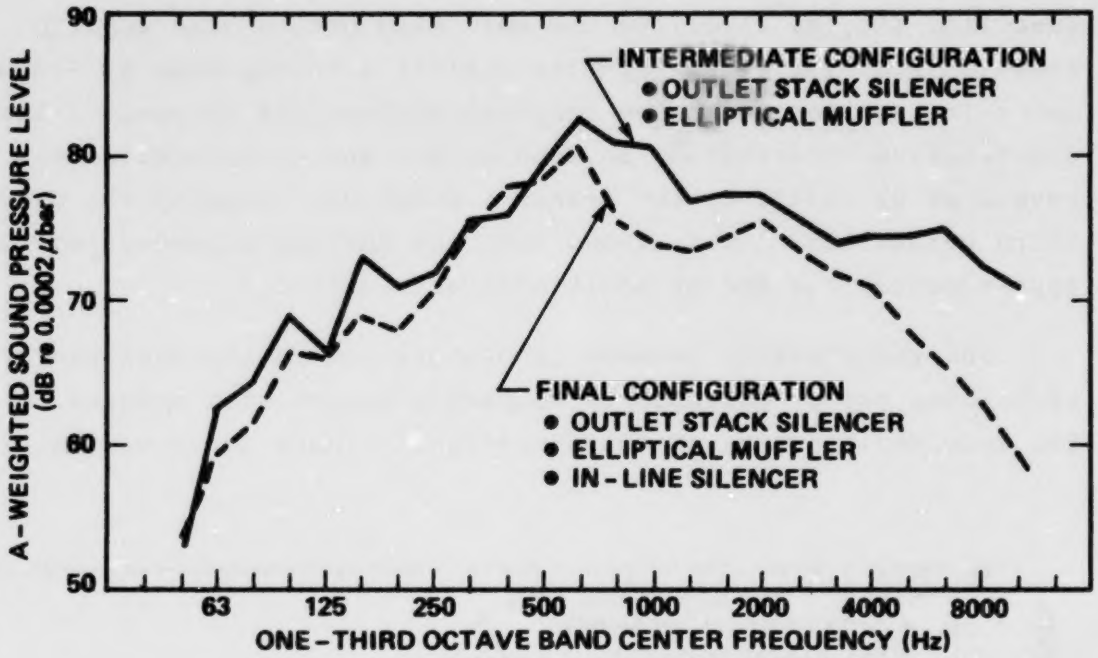

FIG. 15. NOISE LEVELS MEASURED 18 IN. FROM EXHAUST OUTLET DURING ACCELERATION PASSBY TEST.

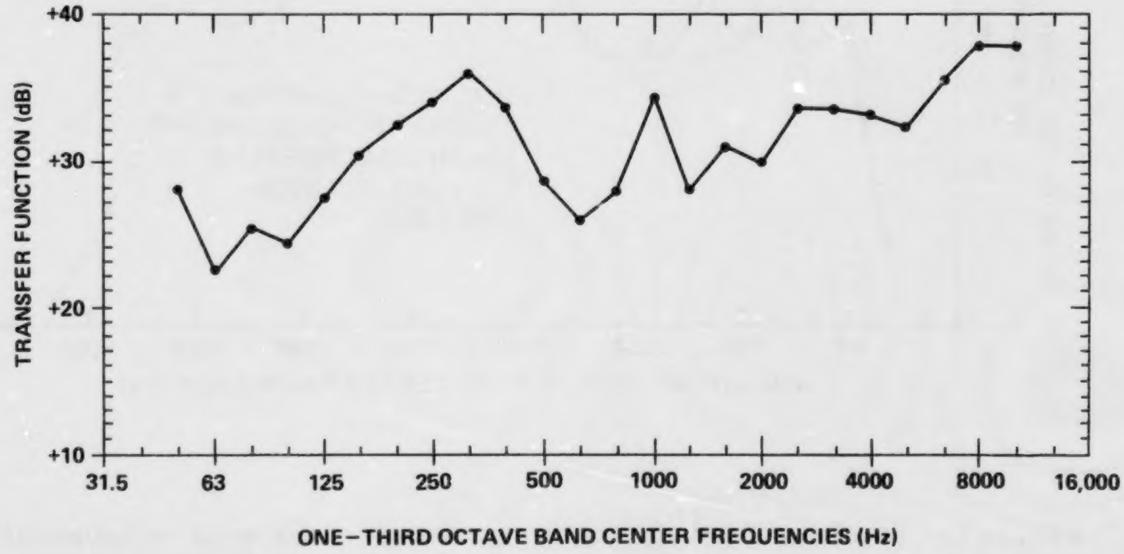

FIG. 16. TRANSFER FUNCTION RELATING NOISE LEVEL AT 18 IN. FROM EXHAUST OUTLET TO LEVEL AT 50 FT.

(see Fig. 15), we calculate the one-third octave band spectra shown in Fig. 17. These spectra exhibit a strong peak at 630 Hz, owing in part to the source spectral content and in part to the constructive interference between direct and ground-reflected waves, as exhibited by the transfer function. Summing the one-third octave band levels shows that the in-line silencer provides approximately 2.6 dBA of additional attenuation.

The relationship between laboratory and passby measurement techniques may be assessed by comparing octave band spectra for the intermediate exhaust configuration. Figure 18 shows that the

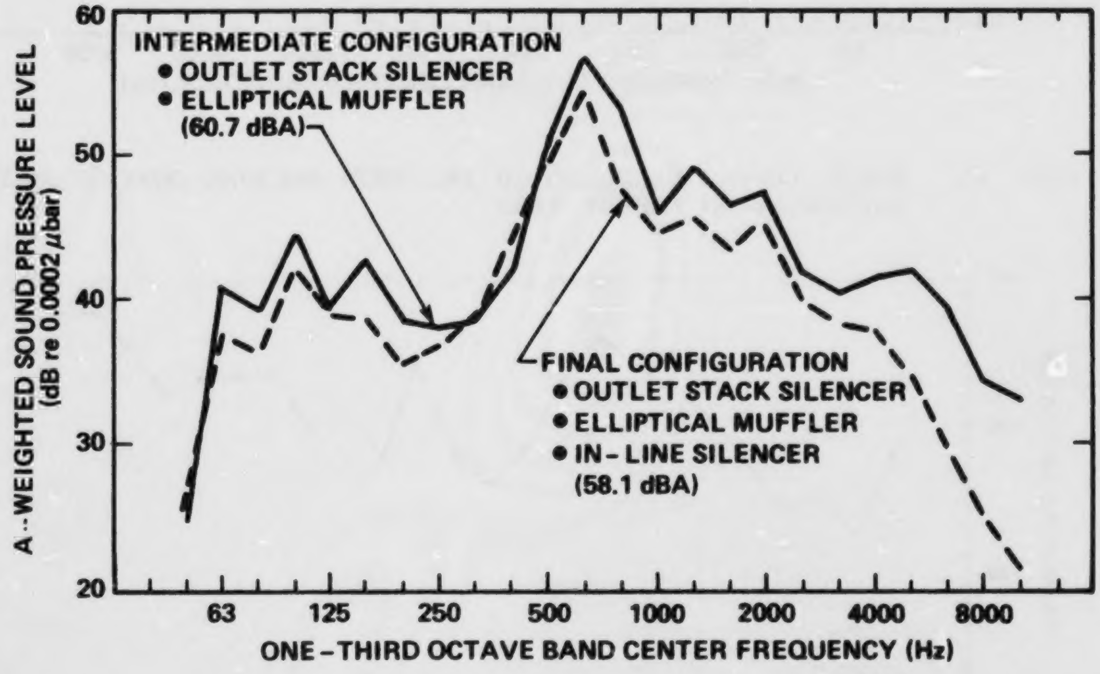

FIG. 17. NOISE SPECTRA AT 50 FT ESTIMATED FROM NEAR MICROPHONE MEASUREMENTS (FIG. 15) AND A TRANSFER FUNCTION (FIG. 16).

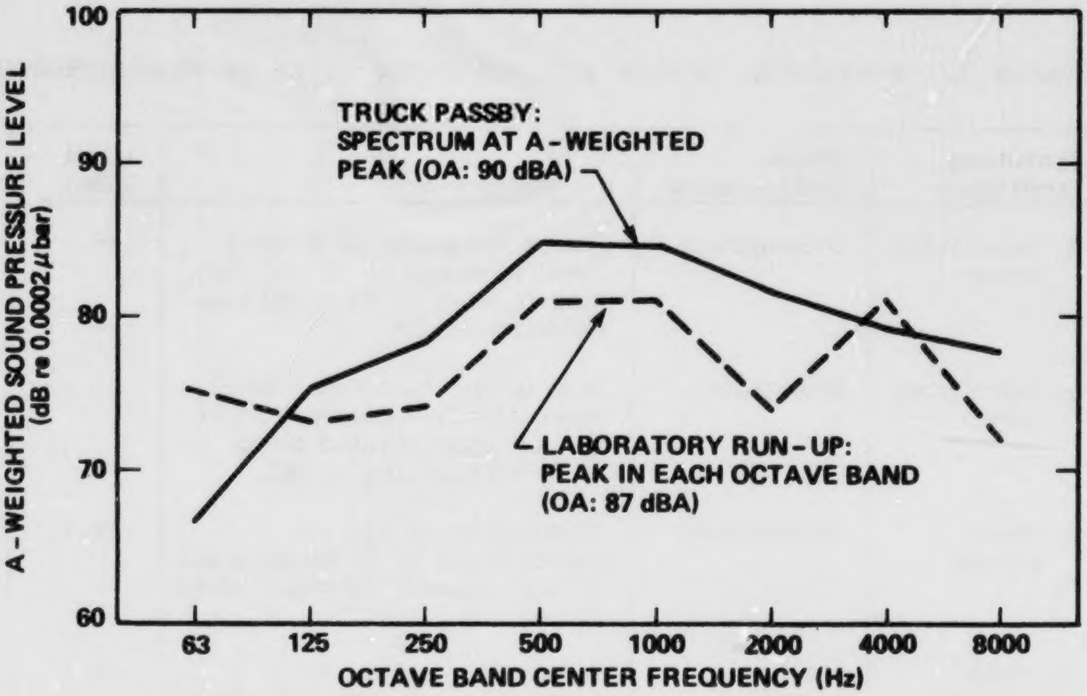

FIG. 18. COMPARISON OF LABORATORY RUNUP AND TRUCK PASSBY LEVELS MEASURED 18 IN. FROM EXHAUST OUTLET.

agreement is within 2 to 8 dB at any frequency; the overall level is 3 dBA higher for the passby measurement.

In summary, the A-weighted levels for the different types of measurements and exhaust system configurations are shown in Table 3. Although both laboratory runup levels are in good agreement with each other, the sum of the peak octave band levels is higher than the other level. This is as expected, because the peaks would occur at different times in the runup cycle and are not strictly additive. For the intermediate configuration, the truck passby level is somewhat higher than the laboratory levels. This could be a result of differences in operating conditions and extrapolation methods, as well as possible contamination of the measured level by other vehicle sources. We shall, of course, use 58.1 dBA as the measure of exhaust noise. It is the only one that corresponds to the final exhaust system configuration, and it was acquired under more germane truck passby conditions.

TABLE 3. A-WEIGHTED LEVELS FOR DIFFERENT TYPES OF MEASUREMENTS.

Operating Conditions	System Configuration	Measurement	Level (dBA)
1. Laboratory runup	Intermediate	Peak A-weighting of graphic level measured at 18 in. and extrapolated to 50 ft by subtracting 30 dBA	56
2. Laboratory runup	Intermediate	Peak octave band sound level meter (fast) - measured at 18 in. and extrapolated to 50 ft by subtracting 30 dBA	57
3. Truck passby	Intermediate	Measured at 18 in. and extrapolated to 50 ft by means of the transfer function given in Fig. 16	60.7
4. Truck passby	Final	Measured at 18 in. and extrapolated to 50 ft by means of the transfer function given in Fig. 16	58.1

3.2 Exhaust Line Shell Noise Control

The following three techniques for reducing exhaust line shell noise were investigated:

- Replacing the single-thickness exhaust pipe with a double-walled pipe of the same outside dimensions
- Enclosing the pipe with a fiberglass blanket and a second pipe of larger diameter
- Installing a silencer in the exhaust line between the turbocharger and muffler.

The double-walled pipe holds promise of reducing shell noise through its greater mass and frictional damping created at the interface of the mutually contacting inner and outer pipe sections. A length of double-walled pipe was fabricated and inserted between the turbocharger and the muffler. The following noise levels were measured in the course of evaluating the effectiveness of the double-walled pipe and the sound radiated by other source/path combinations on the vehicle:

		Noise Level - dBA	
Date	**Exhaust Pipe Configuration**	**Left Side**	**Right Side**
1/8/81	unwrapped	74.8	76.0
1/12/81	wrapped with fiberglass and leaded vinyl	74.9	74.2

The dates are shown to emphasize that the tests were performed at significantly different times, which inevitably introduces more uncertainty into the data than if tests are conducted on the same day. Clearly, intrinsic lack of day-to-day repeatability of the passby test caused the apparent 0.1-dBA rise on the left side of the vehicle when the exhaust pipe was wrapped. The pipe, located on the vehicle's right side, will have little effect on the noise measured on the left side. Moreover, wrapping the pipe will certainly not increase the noise levels.

Bearing in mind the uncertainty of the above data, one may estimate the exhaust line contribution to the right side by logarithmically subtracting 74.2 dBA from 76.0 dBA to obtain an estimated level of 71.3 dBA. That this figure is higher than the estimated value of 65.6 dBA for the untreated exhaust shell (see Sec. 2.2) is probably more a reflection of inaccuracies in the estimating procedure than an accurate measure of the double-walled shell noise. Nevertheless, the results are not encouraging. We were seeking definitive reductions in shell noise and therefore investigated an enclosure for the pipe.

The pipe and its enclosure are illustrated in an assembled and disassembled view in Fig. 19. In the truck, the right-hand

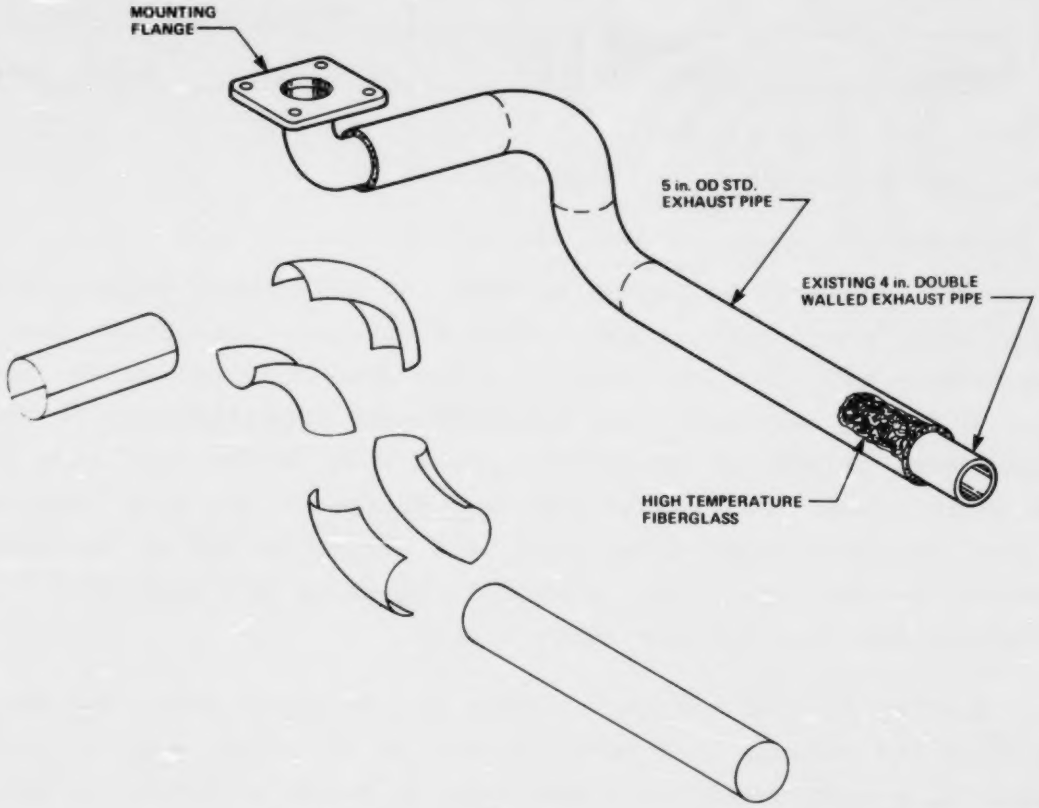
MOUNTING FLANGE
5 in. OD STD. EXHAUST PIPE
EXISTING 4 in. DOUBLE WALLED EXHAUST PIPE
HIGH TEMPERATURE FIBERGLASS

FIG. 19. ENCLOSED EXHAUST PIPE.

end is attached to the turbocharger by means of a length of flex hose, while the flanged end corresponds to the lower flange illustrated in Fig. 11. A half-inch layer of high-temperature fiberglass is used to isolate the outer 5-in. pipe mechanically and acoustically from the existing 4-in. pipe. A straight section of 5-in. pipe is slipped over the right end, while the other three sections are split and reassembled over the remaining 4-in. pipe.

With the exhaust covering held in place by means of seal clamps, vehicle noise levels were measured with the following results:

		Noise Level - dBA	
Date	**Configuration**	**Left Side**	**Right Side**
3/9/81	Enclosed pipe (see Fig. 19)	72.9	73.1

For these tests, enclosure treatment was further developed, primarily through the application of damping material. The overall noise levels were judged satisfactory, but it did not appear that the exterior 5-in. pipe could be welded properly. Misalignments along the lengthwise seams,* the thinness of the metal, and the certainty of melting neighboring fiberglass made the finalization of this solution impractical.

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*The curved sections were made by cutting through 5-in. elbow pieces with a band saw. These elbows contained residual stresses developed during their formation that were partially relieved when the cut was made. The accompanying change in deformation resulted in a failure of two halves to match after they were cut.

The third approach, installing a silencer in the exhaust line, was accomplished by using the same type of 5-in. silencer used for the exhaust stack. Since this silencer is fabricated with relatively thin walls, its contribution to the radiated sound level was evaluated by wrapping it with fiberglass and leaded vinyl. The wrapping proved significant, and a permanent covering made of fiberglass and 5-in. pipe was installed. Noise levels for these configurations are as follows:

		Noise Level - dBA	
Date	**Configuration**	**Left Side**	**Right Side**
7/8/81	Unwrapped	73.7	73.5
7/8/81	Wrapped with fiberglass and leaded vinyl	73.4	72.8
7/31/81	Enclosed with 6-in. pipe	73.2	72.8

A view of the in-line silencer is shown in Fig. 20.

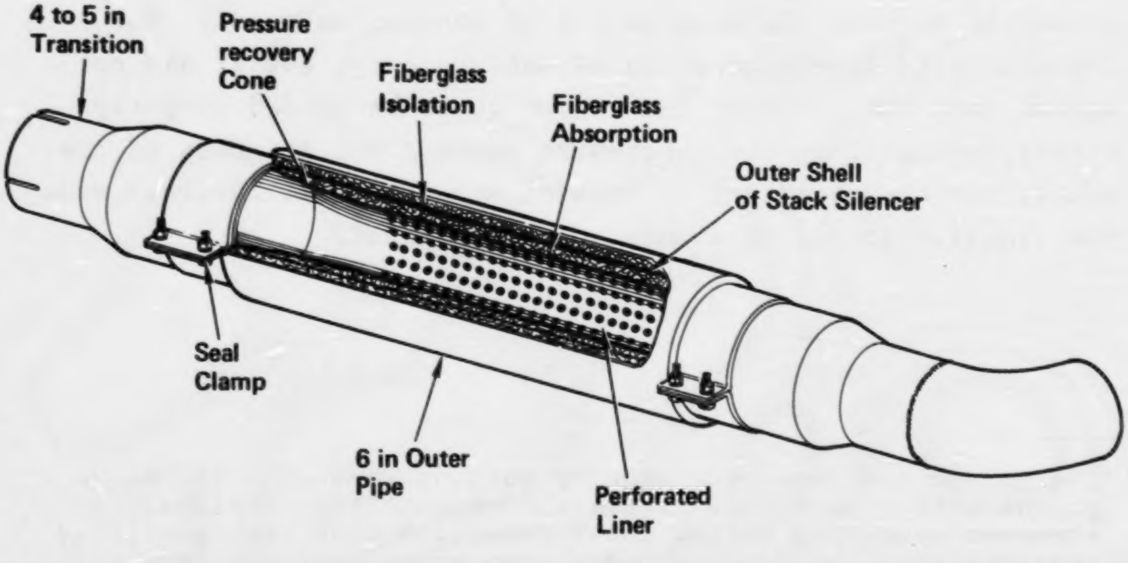

4 to 5 in Transition
Pressure recovery Cone
Fiberglass Isolation
Fiberglass Absorption
Outer Shell of Stack Silencer
Seal Clamp
6 in Outer Pipe
Perforated Liner

FIG. 20. VIEW OF IN-LINE SILENCER.

3.3 Engine/Transmission Airborne Sound Treatment

The baseline contribution of the engine and transmission airborne levels to the overall noise level was estimated to be 79.0 dBA on the left side of the truck and 80.9 dBA on the right side. This source was treated with an acoustic enclosure built around the engine/transmission. Special two-stage engine mounts were installed to control structureborne sound radiation. This treatment is illustrated in Fig. 21, and major components are described in Table 4.

TABLE 4. DESCRIPTION OF ENCLOSURE PANELS.

Designation	Description
L1, R2	Left and right side shields above the frame rail
L2, R2	Left shelf and right shelf above the frame rail and sealing against L1 and R1
L3, R3	Left and right side panels of the bellypan forward of the firewall
L4, R4	Left and right middle side panels of the bellypan between the firewall and the back of the transmission
L5, R5	Left and right rear side panels from rear of transmission to 3 ft aft of the cab
B1, B2, B3	Panels forming the bottom of the bellypan
B4, B5	Panel B2 is fixed, the others are held by quick release fasteners

The following overall design objectives guided the design of the enclosure:

- Adequate noise reduction
- Minimal effect on engine cooling performance
- Minimal maintenance interference
- Simplicity and ease of construction

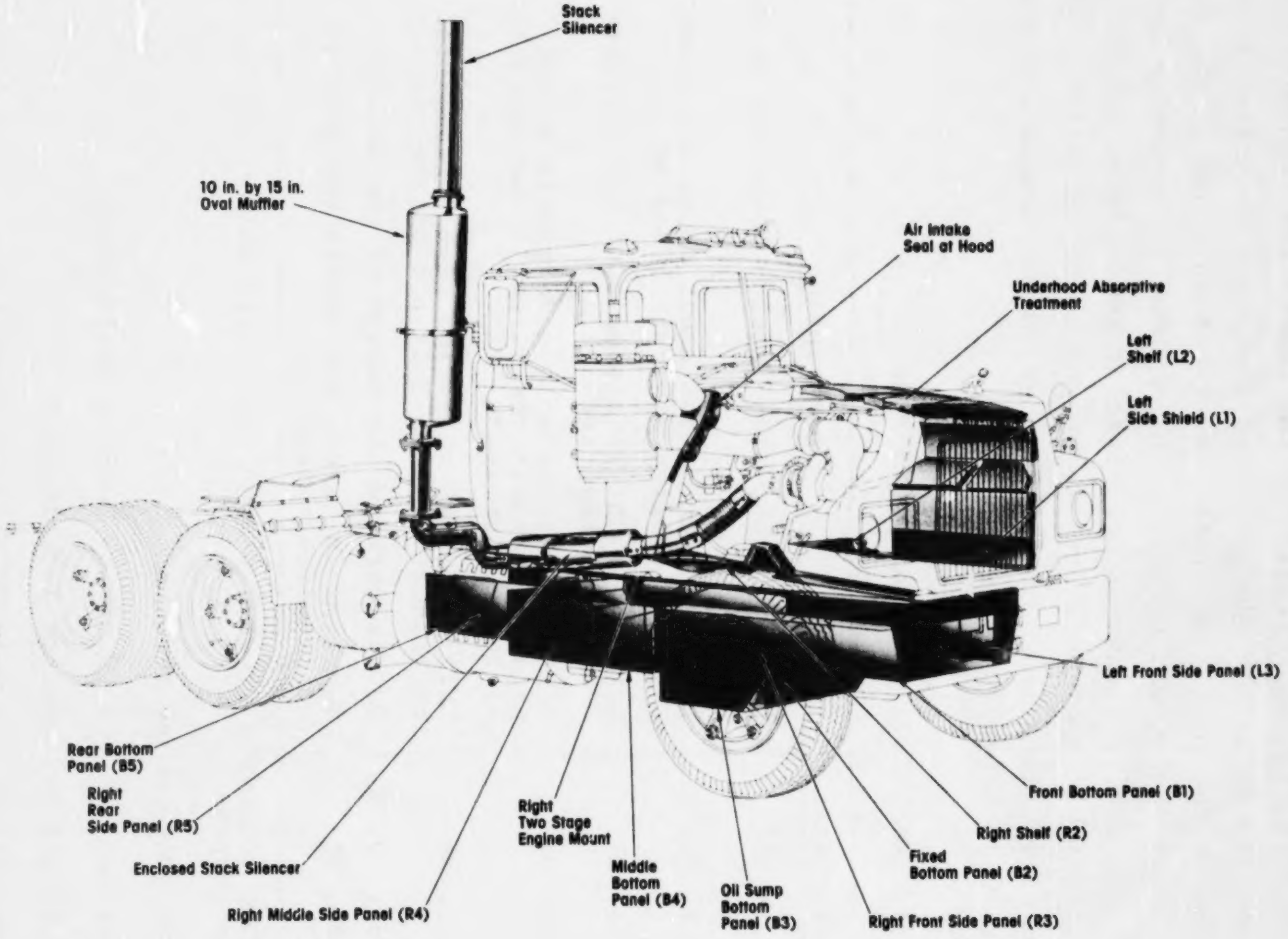

Stack Silencer
10 in. by 15 in. Oval Muffler
Air Intake Seal at Hood
Underhood Absorptive Treatment
Left Shelf (L2)
Left Side Shield (L1)
Left Front Side Panel (L3)
Rear Bottom Panel (B5)
Right Rear Side Panel (R5)
Enclosed Stack Silencer
Right Middle Side Panel (R4)
Right Two Stage Engine Mount
Middle Bottom Panel (B4)
Oil Sump Bottom Panel (B3)
Right Front Side Panel (R3)
Fixed Bottom Panel (B2)
Right Shelf (R2)
Front Bottom Panel (B1)

FIG. 21. THE MACK R686 WITH NOISE CONTROL TREATMENTS.

- Durability
- Protection of sound-absorptive material from environmental contaminants
- Light weight.

Enclosure Design Concept

A tunnel-like enclosure was designed to shield the community from engine and transmission noise. The enclosure is open at the front and rear of the truck to allow cooling air to flow through the radiator, over the engine and transmission, and out the rear. The hood and the bottom of the cab form part of the top of the enclosure, as illustrated in Fig. 21. Aft of the back of the cab, the top of the enclosure is formed by 0.19-in.-thick aluminum diamond plate deck installed as part of this program. The remaining major areas requiring treatment to complete the enclosure are:

- The area between each frame rail and the inner fenders of the fiberglass hood
- The area between each frame rail and the bottom of the cab
- The area beneath the engine and between the frame rails
- The area on each side of and beneath the drive shaft below the diamond plate decking aft of the cab.

The Mack R686 came equipped with virtually no noise control treatments in the engine compartment, although the inner fenders of the fiberglass hood did partially block the line of sight from the roadside through the wheel wells to the engine, as illustrated earlier in Fig. 5. That treatment was not adequate for the level of engine-noise reduction required here. Consequently, the inner fenders were modified with panels L1 and R1. In addition, the frame rails were extended with panels L2 and R2, which, together with the walls of the inner fenders, seal the space between the hood and the frame rail from the radiator to the firewall.

Below the frame rails, panels L3 and R3 form the sidewalls of the bellypan forward of the firewall. From the firewall to the rear of the transmission, panels L4 and R4 perform the same function as do panels L5 and R5 from the rear of the transmission to 3 ft aft of the cab. Panels B1, B2, B3, B4, and B5 close the bottom of the bellypan from the radiator to the back of the cab.

The gaps between the bottom of the cab and the frame rails are sealed with 0.160-in. aluminum panels and 1/8-in.-thick rubber sheets. These gap shields extend from the firewall to the back of the cab on both sides of the vehicle.

Except as noted above, the enclosure is fabricated primarily from sheet aluminum. While it is anticipated that a truck manufacturer would use an alternative material (e.g., sheet steel), sheet aluminum provides a light, rigid material well suited to prototype work. A minimum panel thickness of 1/8 in. was dictated by requirements for strength and durability rather than for noise reduction. This 1/8-in. aluminum panel thickness is more than adequate to provide the required noise reduction [3].

Sound-Absorptive Material

Two types of absorptive treatments were used in the enclosure:

- BBN-installed 1.5-in. Mylar-wrapped fiberglass behind perforated aluminum sheet metal
- BBN-installed 2-in. unprotected fiberglass.

The 1.5-in. Mylar-wrapped fiberglass was attached to panels L4, R4, L5, and R5 from the front of the transmission to the rear of the enclosure below the frame rails. Figure 22 shows the absorptive treatment on panels L5 and L4. This type of absorptive treatment and its acoustic performance have already been described elsewhere [3].

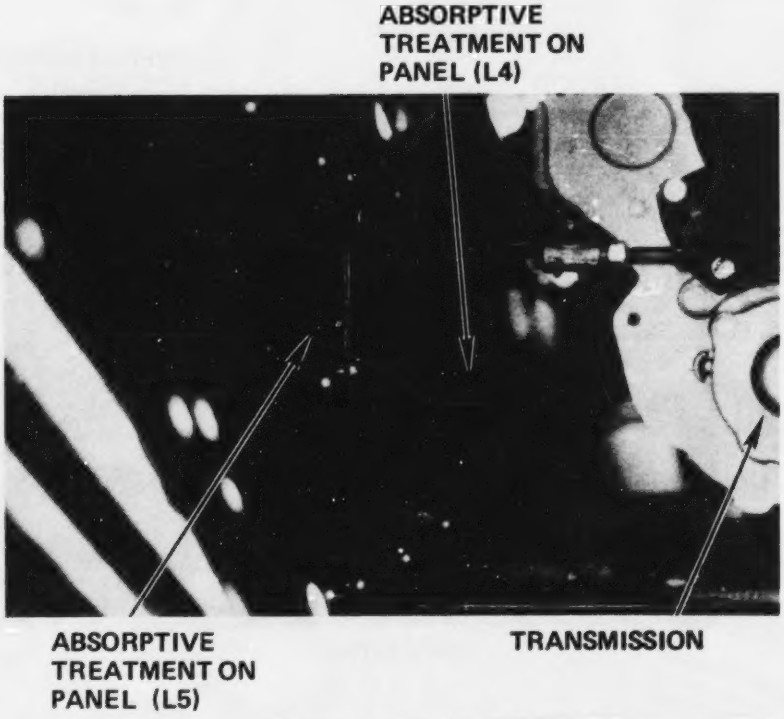

ABSORPTIVE TREATMENT ON PANEL (L4)
ABSORPTIVE TREATMENT ON PANEL (L5)
TRANSMISSION

FIG. 22. ABSORPTIVE TREATMENT ON LEFT SIDE OF ENCLOSURE, FROM REAR OPENING LOOKING FORWARD.

The 2-in.-thick unprotected fiberglass is installed on the inner surface of the hood above the frame rails (Fig. 23), and on the underside of the cab floor above the transmission. These are areas that, because of their remoteness, are unlikely to receive much mechanical damage. In addition, they tend to be high up in the enclosure where contamination by water and oil is less of a problem. Accordingly, it was decided to forego the use of perforated metal for mechanical protection and the use of Mylar wrapping to prevent contamination in these areas.

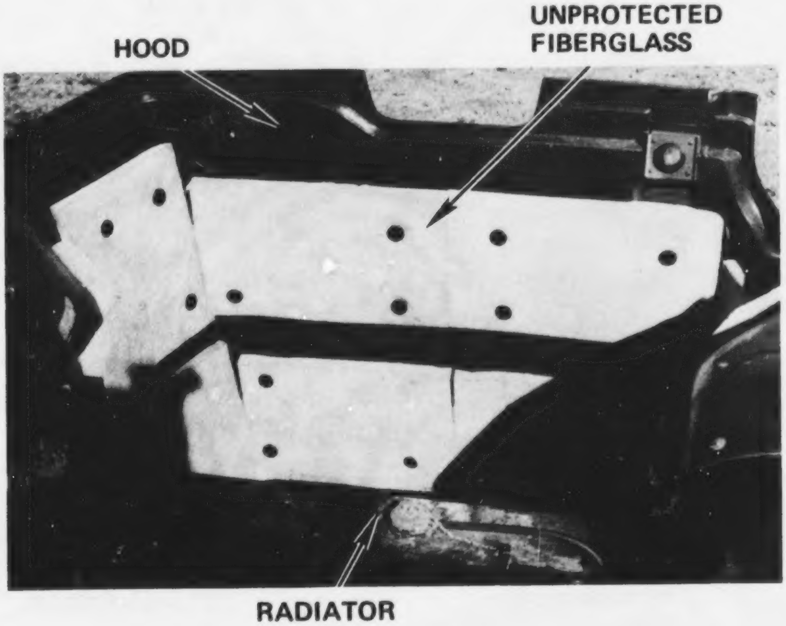


FIG. 23. UNPROTECTED FIBERGLASS ON INNER SURFACE OF HOOD, LOOKING FORWARD FROM CAB WITH HOOD RAISED.

Special Seals

The engine/transmission airborne noise contribution is sufficiently high that special attention was required to seal as completely as is practical all of the enclosure openings other than the opening at the radiator and at the rear of the enclosure to allow for the passage of cooling air. Two of the seals required to improve the insertion loss, the hood seal and air intake seal, are described here. Other seals will be described as we discuss the various elements of the enclosure.

As originally equipped, the Mack had no seal where the hood joins the cab. Since there were significant gaps at a number of locations along this junction, we installed a foam rubber strip, as shown in Fig. 24. The strip was originally approximately 1 in. square. It was shaved as necessary to obtain a good seal along the full length of the hood/cab junction.

The air cleaner on the Mack is external to the engine compartment (see Figs. 4 and 21). As originally configured, the air intake and air-to-air intercooler ducts passed through an opening in the hood. That opening was sealed as shown in Fig. 25 by fashioning a rubber boot to fit tightly around the ducts and seal against the hood opening.

Side Shields

Two side shields were added to the inner fenders, and two side shelves were attached to the frame rail so that an airtight seal could be formed between the hood and frame rails from the firewall to the radiator. Figure 26 shows the right side shield (R1) with the hood raised. It is a very short aluminum panel bolted to the front of the inner fender enabling that panel to form a seal with a foam rubber gasket (also shown in the figure) that is glued to the right side shelf (R2).

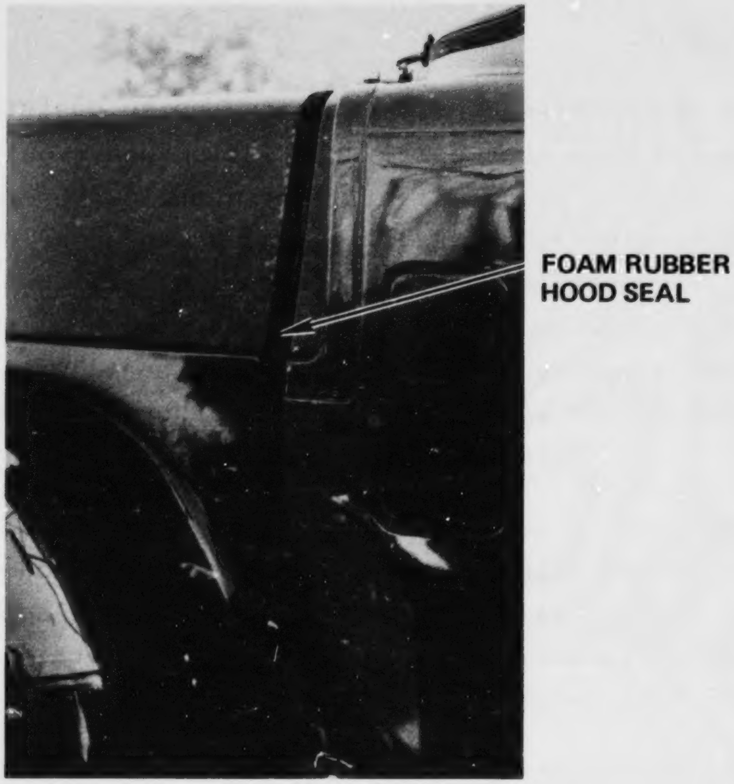
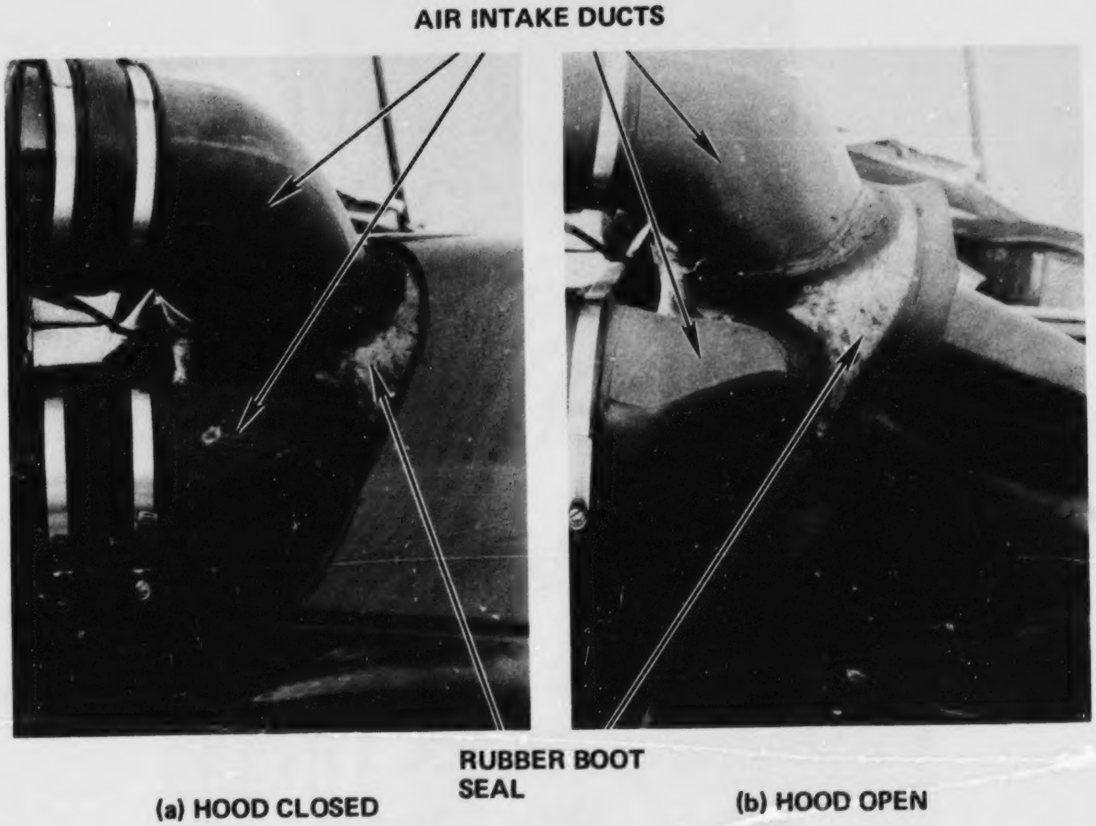


FIG. 24. HOOD SEAL AS SEEN FROM LEFT SIDE OF TRUCK.

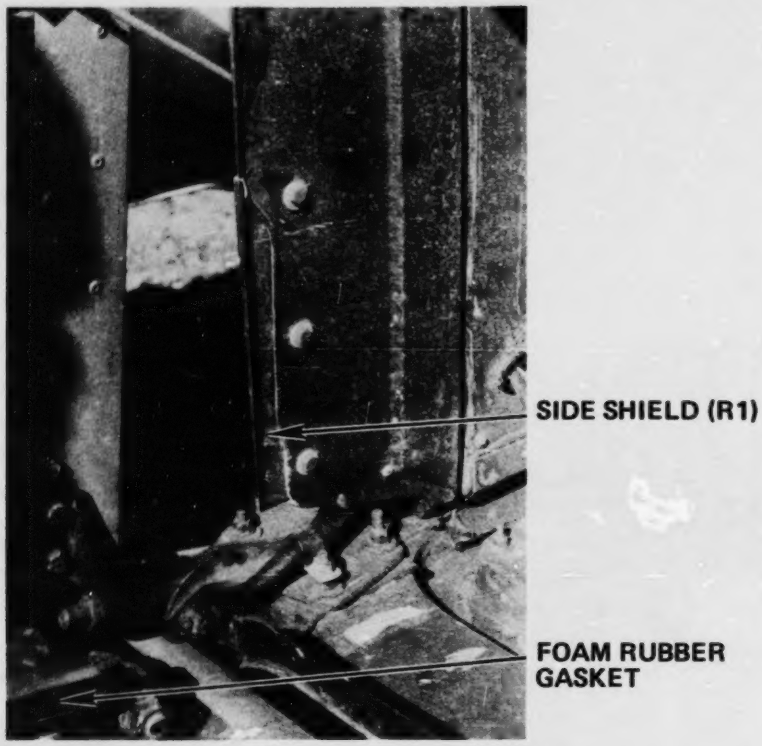
The right side shelf is made up of two panels. The forward portion (R2F) attaches directly to the top surface of the frame rail, extending from just aft of the shock absorber to the radiator. Figure 27 shows that panel with the hood open. The foam rubber gasket running the full length of the shelf is also shown in the figure. The rear portion of the right side shelf (R2A), shown in Fig. 28, seals the space between the rear of the fender, the cab, and the frame rail. A good seal with the hood is achieved by means of a rubber flap that "wipes" the back of the fender just above the mud flap. The rear of this shelf has an opening to allow for the passage of the exhaust pipe.


AIR INTAKE DUCTS
RUBBER BOOT SEAL

(a) HOOD CLOSED

(b) HOOD OPEN

FIG. 25. AIR INTAKE SEAL AS SEEN FROM RIGHT SIDE OF TRUCK.


SIDE SHIELD (R1)
FOAM RUBBER GASKET

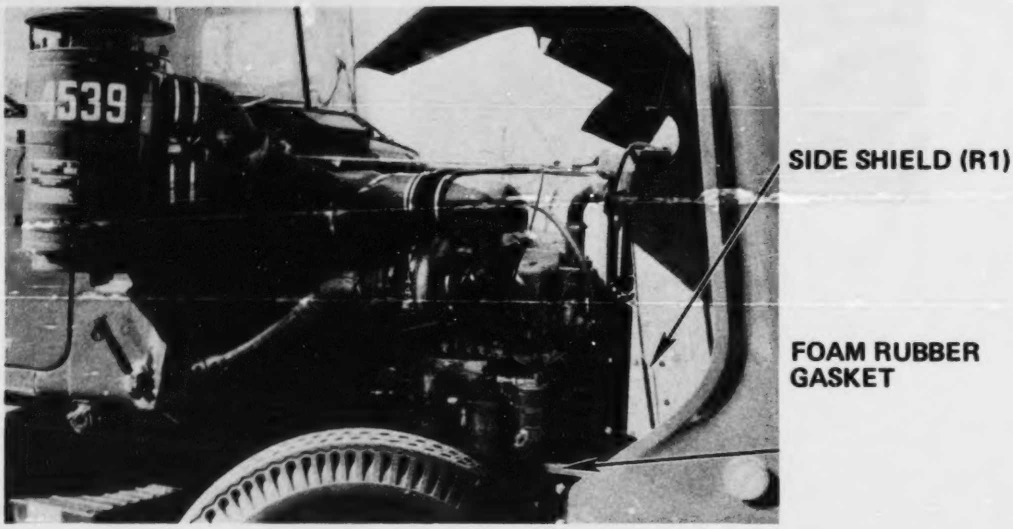

4539
SIDE SHIELD (R1)
FOAM RUBBER GASKET

FIG. 26. RIGHT SIDE SHIELD AS SEEN FROM RIGHT OF TRUCK WITH HOOD RAISED.

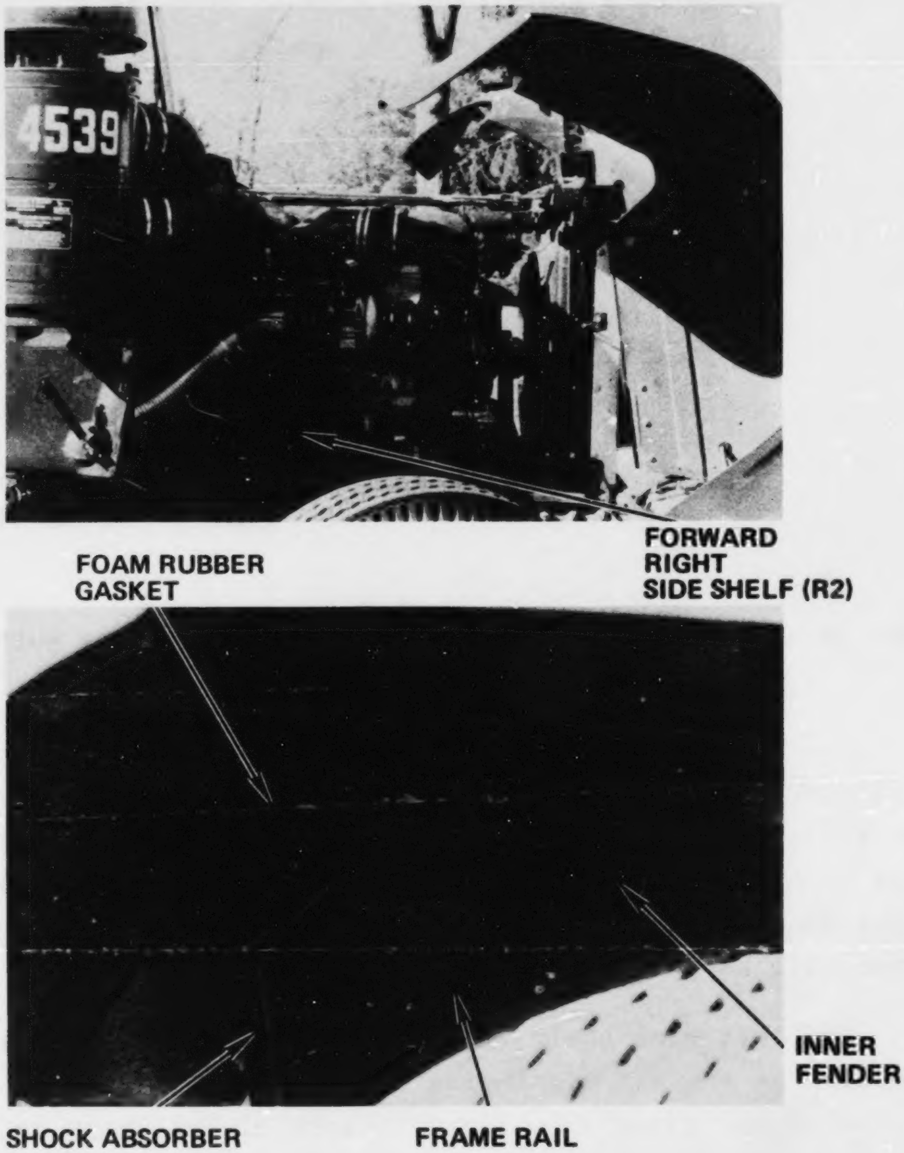

4539
FORWARD RIGHT SIDE SHELF (R2)
FOAM RUBBER GASKET
INNER FENDER
SHOCK ABSORBER
FRAME RAIL

FIG. 27. RIGHT SIDE SHELF AS SEEN FROM RIGHT SIDE OF TRUCK.

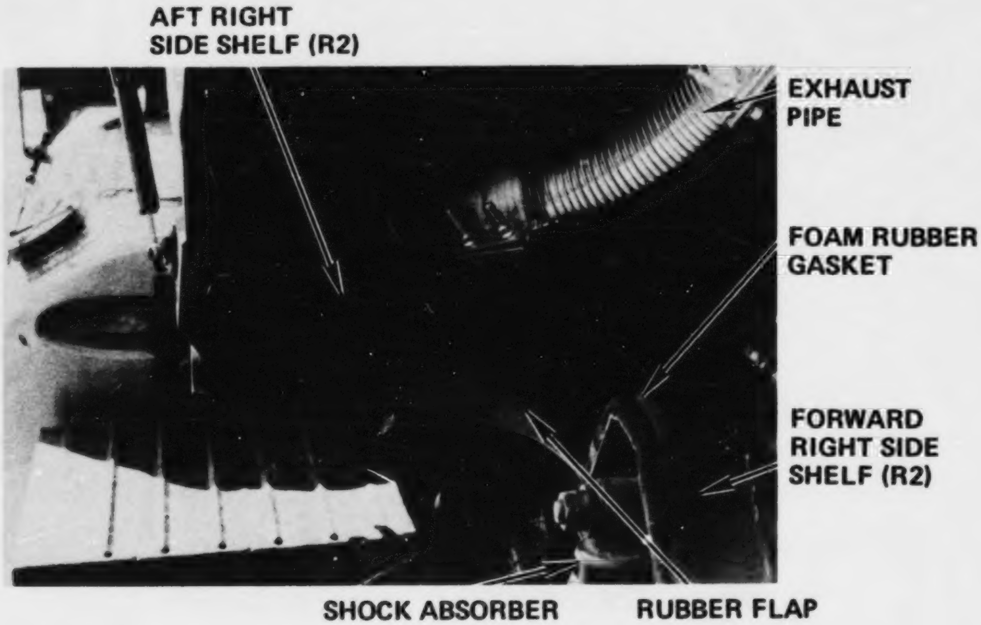

AFT RIGHT
SIDE SHELF (R2)
EXHAUST
PIPE
FOAM RUBBER
GASKET
FORWARD
RIGHT SIDE
SHELF (R2)
SHOCK ABSORBER
RUBBER FLAP

FIG. 28. RIGHT SIDE SHELF, LOOKING AFT FROM RIGHT WHEEL WELL WITH HOOD RAISED.

The left side shield (L1) is shown in Fig. 29. It is an aluminum panel bolted to the left inner fender and extending from the radiator to just aft of the shock absorber. On its lower edge is a foam rubber gasket, the same material as used on the right side shelf (R2), that seals against the top flange of the frame rail as shown in Fig. 30.

The left side shelf (L2) is similar to and peforms the same function as the aft right side shelf (R2). It is shown in Fig. 31. A rubber P-seal, so named because of its shape in cross section, and a rubber flap seal against the back side of the left fender just above the mud flap.

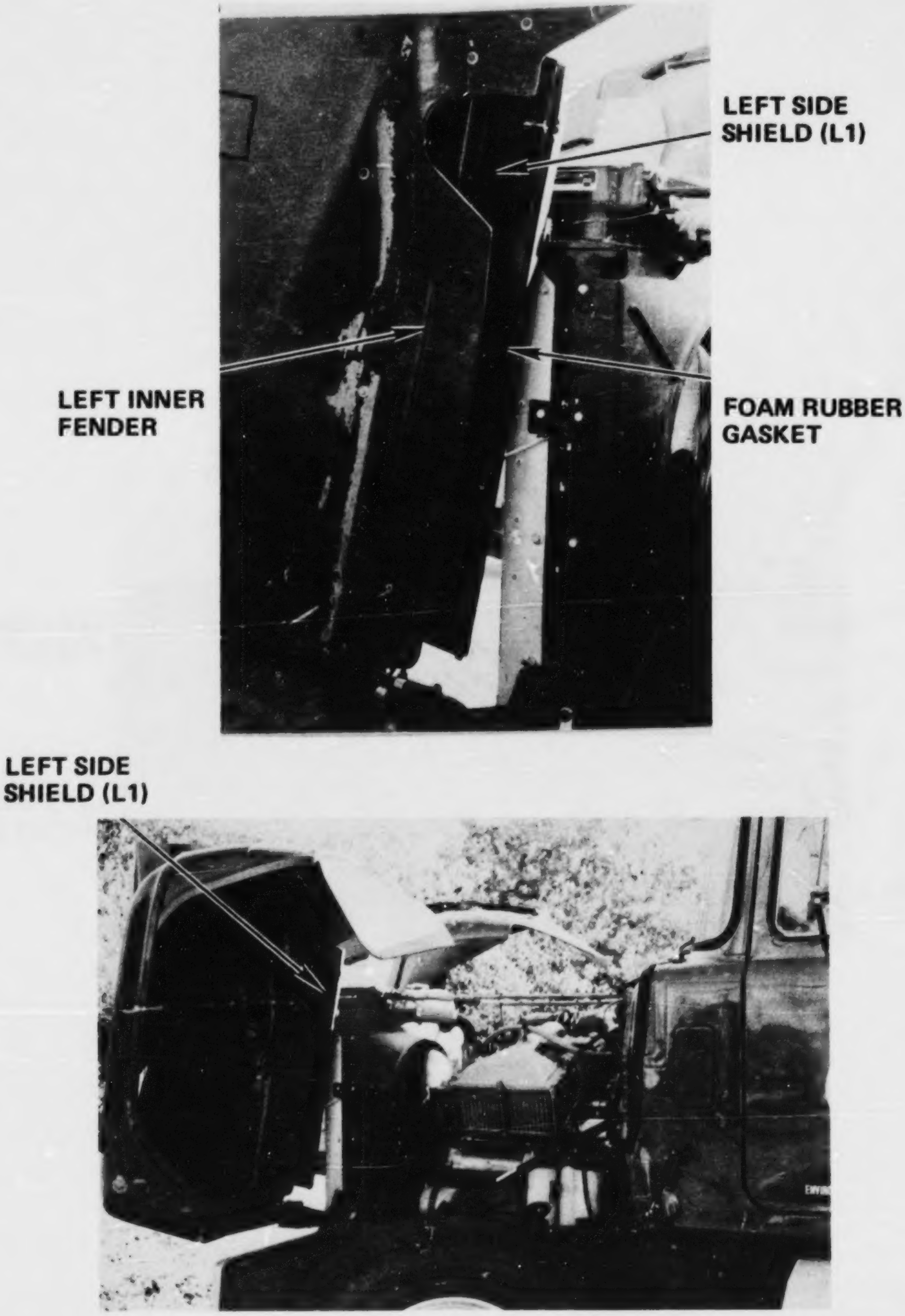

LEFT SIDE SHIELD (L1)
LEFT INNER FENDER
FOAM RUBBER GASKET
LEFT SIDE SHIELD (L1)

FIG. 29. LEFT SIDE SHIELD AS SEEN FROM LEFT SIDE OF TRUCK.

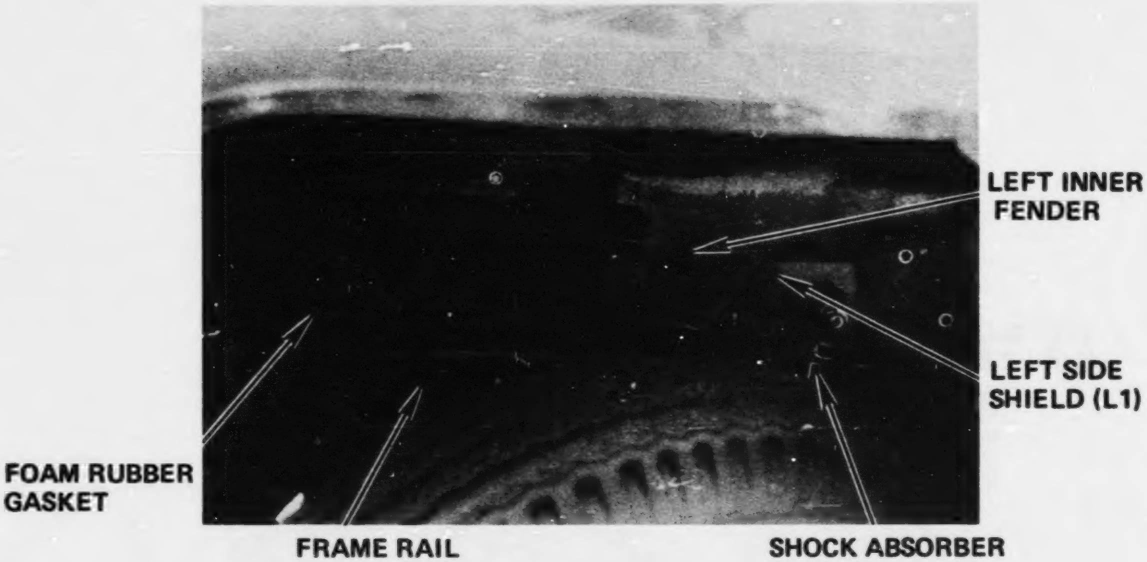

LEFT INNER FENDER
LEFT SIDE SHIELD (L1)
FOAM RUBBER GASKET
FRAME RAIL
SHOCK ABSORBER

FIG. 30. LEFT SIDE SHIELD (L1) AS SEEN FROM LEFT WHEEL WELL WITH HOOD CLOSED.

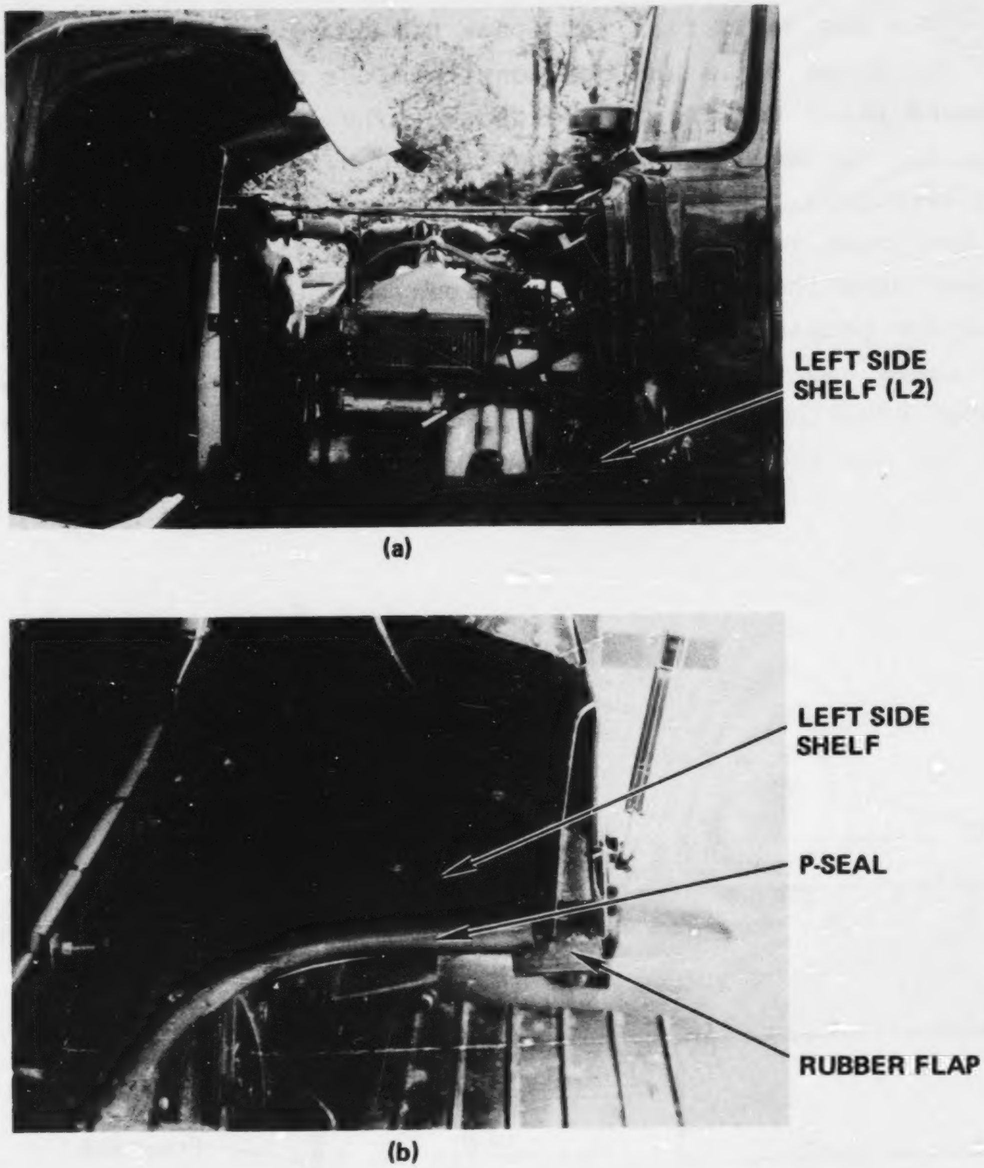

LEFT SIDE SHELF (L2)
(a)
LEFT SIDE SHELF
P-SEAL
RUBBER FLAP
(b)

FIG. 31. LEFT SIDE SHELF (L2): (a) AS SEEN FROM LEFT SIDE OF TRUCK AND (b) FROM LEFT WHEEL WELL, LOOKING AFT.

Gap Seals

The gap seals fill the space between the floor of the cab and the frame rails and between the floor of the cab and the diamond plate decking at the rear of the cab, as illustrated in Fig. 32. Between the frame rail and cab, the seals are made from two materials. A 0.160-in. aluminum strip is bolted to the web of the frame rail and extends part way to the floor of the cab. Rubber sheeting 1/8-in. thick bolted to the top edge of the aluminum panels seals the remaining gap between the top edge of the panel and the floor cab. At the rear of the cab, this same rubber sheeting is bolted to the cab body to seal the gap between the cab and the diamond plate deck behind the cab.

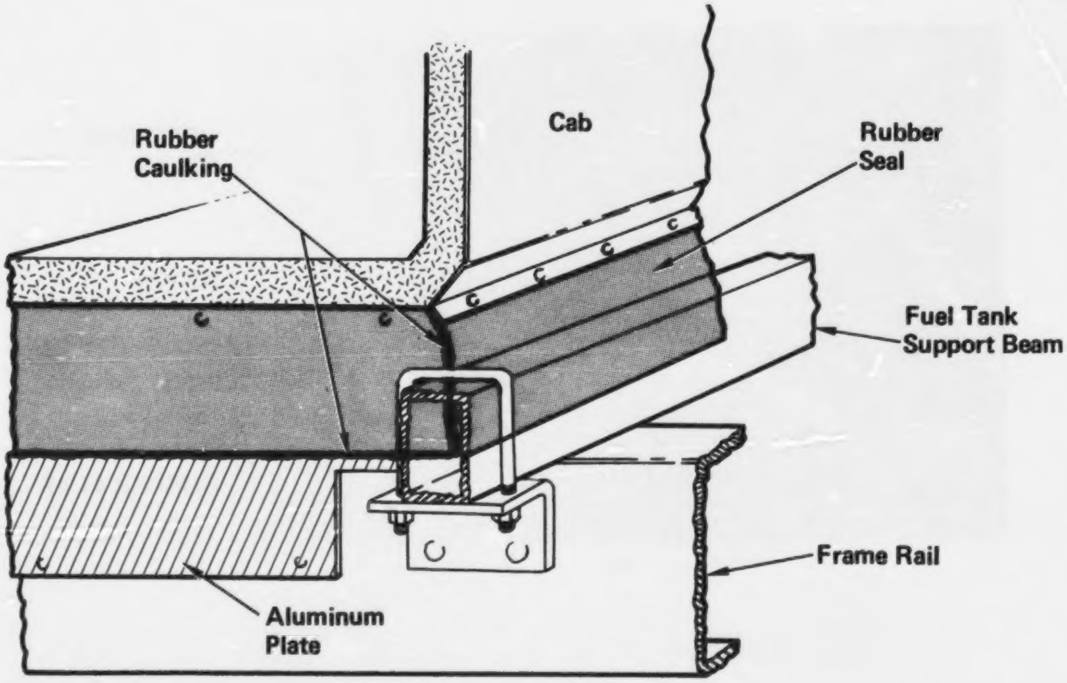

Cab
Rubber Caulking
Rubber Seal
Fuel Tank Support Beam
Frame Rail
Aluminum Plate

FIG. 32. CAB GAP SHIELD.

Bellypan (R3-R5, L3-L5, B1-B5)

The bellypan encloses the bottom of the engine and transmission, extending from the bottom of the radiator to a point 3 ft behind the rear of the cab. The design goals for the bellypan were:

- Maximum accessibility for maintenance purposes
- No reduction of ground clearance
- Quick removal and replacement of bottom panels
- Provision for drainage
- Adequate clearance over front axle.

All the side panels (R3 through R5 and L3 through L5) are fabricated from 0.160-in. aluminum. The panels, which are attached to the frame rails with brackets, start at the bottom flange of the frame rail and extend down to form the side walls of the bellypan.

Just aft of the transmission, the enclosure narrows from the right side (panel R5) to avoid interference with the pump and power takeoff (PTO) unit in that area. Figure 33 shows the opening in the panel (R5) to allow passage of the PTO shaft from the transmission to the pump.

The bottom of the bellypan is sealed with five panels, all fabricated from 0.125-in. aluminum. Panels B1, B3, B4, and B5 are attached to the side panels with quick-release quarter-turn fasteners (Southco Model No. 85). The panels are designed to be removed and reinstalled quickly and easily for routine maintenance of the engine and transmission. The remaining panel, the fixed bottom panel (B2), shown in Fig. 21, is bolted to side panels R3 and L3 to add rigidity to the enclosure. Figures 34, 35, and 36 provide a number of views of the bellypan as installed on the truck.

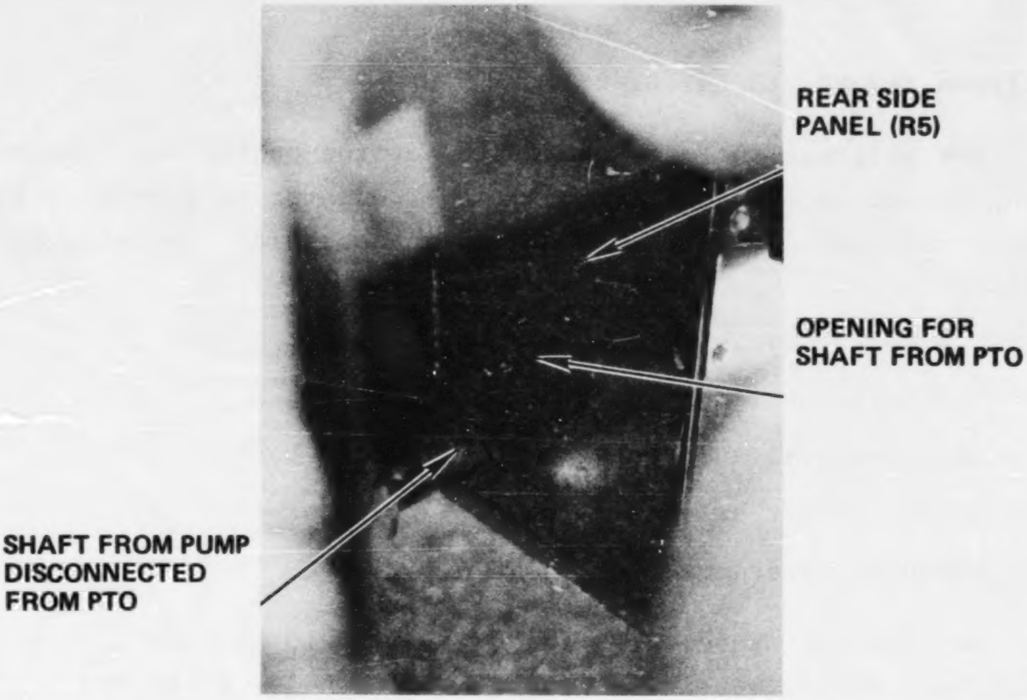

REAR SIDE PANEL (R5)
OPENING FOR SHAFT FROM PTO
SHAFT FROM PUMP DISCONNECTED FROM PTO

FIG. 33. OPENING IN PANEL R5 FOR PTO SHAFT.

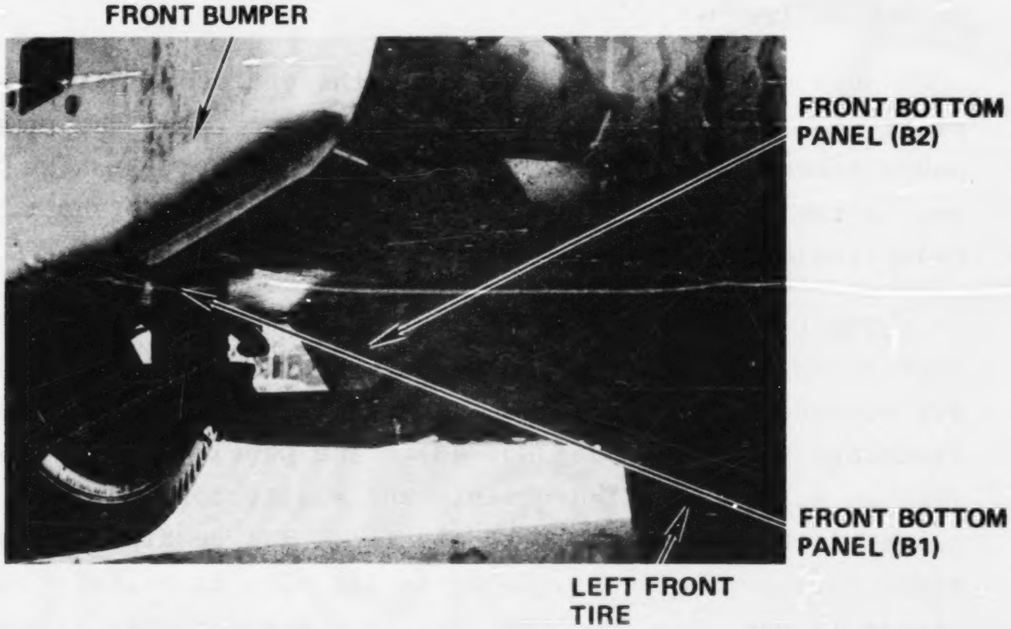

FRONT BUMPER
FRONT BOTTOM PANEL (B2)
FRONT BOTTOM PANEL (B1)
LEFT FRONT TIRE

FIG. 34. FORWARD PORTION OF BELLYPAN AS SEEN FROM BENEATH LEFT SIDE OF BUMPER.

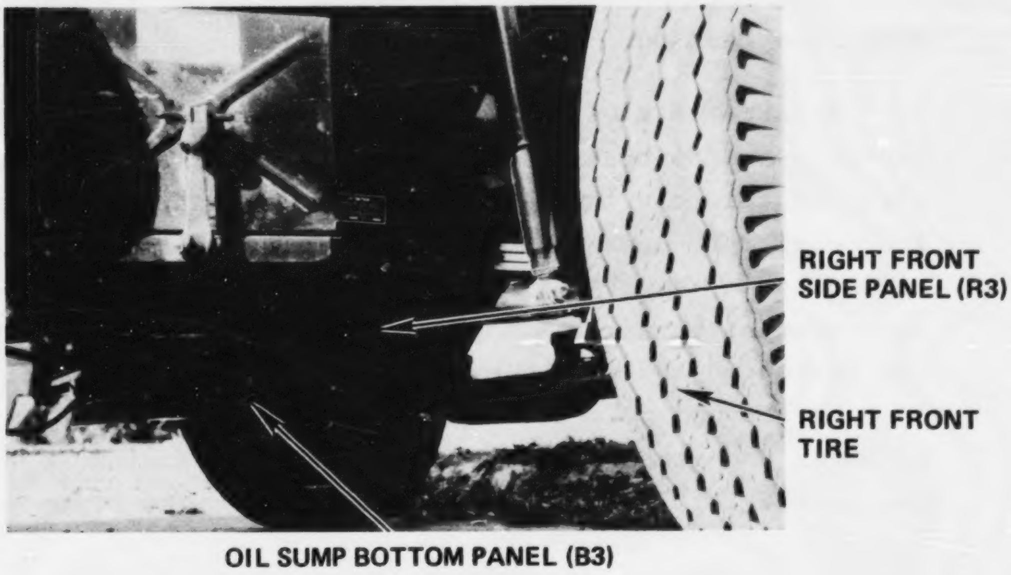

RIGHT FRONT SIDE PANEL (R3)
RIGHT FRONT TIRE
OIL SUMP BOTTOM PANEL (B3)

FIG. 35. BELLYPAN IN VICINITY OF OIL SUMP AS SEEN FROM BEHIND RIGHT FRONT TIRE.

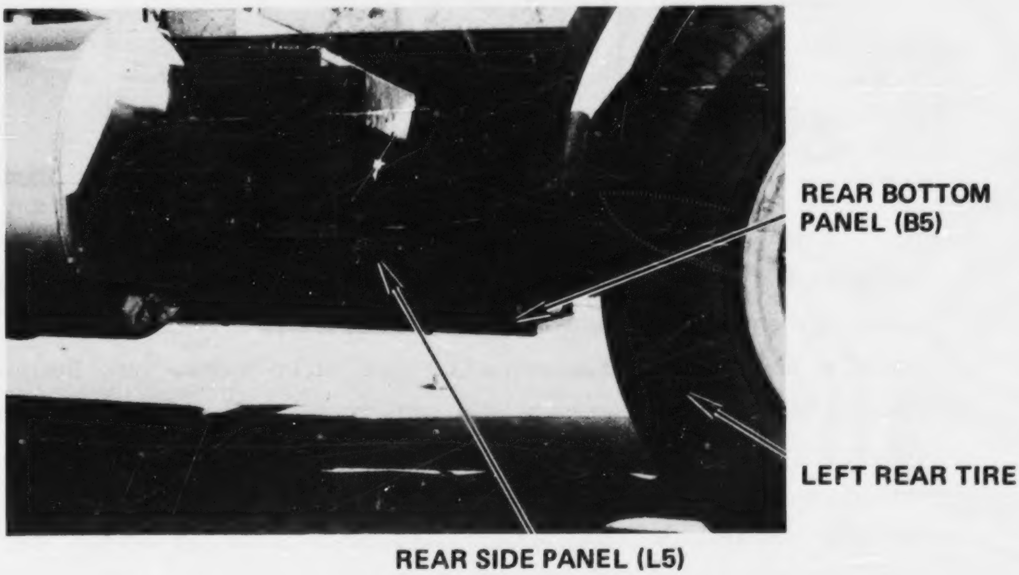

REAR BOTTOM PANEL (B5)
LEFT REAR TIRE
REAR SIDE PANEL (L5)

FIG. 36. REAR OF BELLYPAN AS SEEN FROM JUST FORWARD OF LEFT REAR TIRE.

Enclosure Damping

Measurements of noise radiated from aluminum enclosure panels and of vibration on the panels indicated that the enclosure was transmitting significant levels of structureborne sound. The results of measurements of the noise from the truck operated according to the EPA acceleration test procedure [7] with the enclosure alternately wrapped with leaded vinyl and fiberglass and unwrapped are shown in Table 5. By logarithmically subtracting the wrapped noise levels from the unwrapped, we estimated the airborne and structureborne noise from the enclosure to be 69.7 dBA on the left side and 65.8 dBA on the right side.

TABLE 5. TRUCK NOISE WITH THE ENCLOSURE WRAPPED AND UNWRAPPED.

	Left Side			Right Side		
	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs
Enclosure wrapped	72.9	0.3	6	73.4	0.2	4
Enclosure unwrapped	74.6	0.5	4	74.1	0.5	5

Vibration levels measured on the aluminum enclosure are compared in Fig. 37 with measurements at the same location on a mock-up enclosure made of Masonite. The Masonite and aluminum panels have a similar density per unit area, but because the Masonite panels have higher internal losses, the almost 10-dB higher panel vibration levels indicated the potential need for damping. Measurements of the loss factor on the aluminum panels of the enclosure indicated that some benefit might be realized by additional damping. Table 6 shows some typical values of the

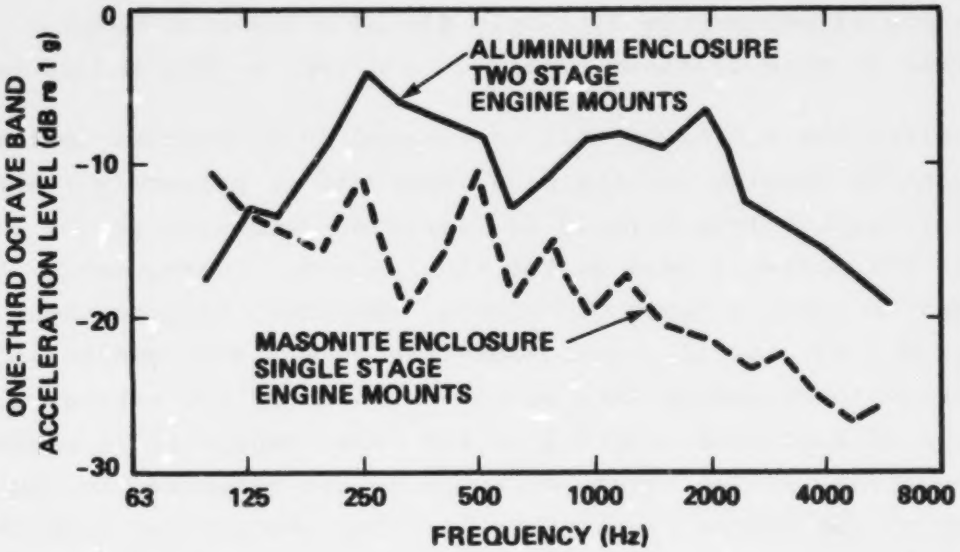


FIG. 37. COMPARISON OF MASONITE AND ALUMINUM ENCLOSURE PANEL VIBRATION WITH THE TRUCK OPERATED ACCORDING TO EPA TEST PROCEDURE - PANEL R3 OPPOSITE THE FRONT AXLE.

TABLE 6. LOSS FACTORS MEASURED ON PANELS OF ALUMINUM ENCLOSURE (%).

Frequency	Panel						
	B1	B3	B4	B5	R3	R4	R5
63	11.0	4.2	-	3.0	-	-	7.4
125	1.7	3.9	1.6	0.84	9.8	-	8.8
250	1.3	2.5	1.0	0.40	5.9	4.7	7.8
500	2.9	1.7	1.0	0.67	3.2	5.2	2.6
1000	1.5	2.1	0.81	0.39	3.1	3.3	1.8
2000	0.73	1.5	0.60	0.32	3.1	2.9	1.5
4000	0.65	1.1	0.90	0.60	2.4	1.9	1.7
8000	0.97	0.92	0.46	0.46	1.2	1.1	0.92

loss factors for the enclosure panels. Ten percent loss factor is generally achievable by simply gluing a layer of damping material to thin aluminum panels of the type in the enclosure.

Since the vibration will be reduced in proportion to the increase in damping (if the panel response is primarily resonant) we would expect from 5 to 10 dB vibration reduction on the bottom panels and somewhat less on the side panels. Consequently, we decided to apply a layer of 3/16-in. EAR C2003 damping sheet to panels B1, R3, and L3, since vibration levels were generally higher on those panels than on other panels of the enclosure. We planned to apply the material to the other panels if it seemed to be required. In the course of applying the material, we found points on the bottom panels where casting projections from the transmission housing were apparently contacting the bottom panels, as paint had been rubbed away at those locations. These projections were ground away, and the truck was tested with the damping material applied to the three enclosure panels described above. Appendix C shows that with the truck in this configuration, the noise was reduced to 72.9 dBA on the left and 73.0 dBA on the right - essentially the same noise levels as with the enclosure wrapped. Unfortunately, we do not know which treatment, the damping or the removal of the potential flanking paths to the enclosure, caused the observed elimination of noise from the enclosure as a significant contributor to noise from the truck. Since the problem was solved, removing the damping and remeasuring the noise never became a priority item.

3.4 Engine/Transmission Structureborne Sound Treatment

Early in the program we found that structureborne vibration from the engine and transmission, while not a major noise source in the untreated Mack R686, could be a significant contributor after exhaust noise and engine/transmission airborne noise were reduced. The primary contribution was at 500 Hz, a frequency associated with the tooth passage frequency of a pair of gears in the transmission.

Past experience has shown that significant reduction in engine/transmission structureborne noise from heavy-duty diesel trucks can usually be obtained by improving only the two rear engine mounts [1]. The approach chosen to decrease the transmission of vibration through these mounts was to convert them from single-stage mounts to two-stage mounts. As illustrated schematically in Fig. 38, a two-stage mount incorporates a blocking mass between isolators. If the single-stage mount has been properly designed, such that its deflection under dynamic load is large compared to the deflection of the frame rail at the mounting point, then the insertion loss due to the use of a two-stage mount can be readily calculated. The calculation shows that the increase in vibration isolation is given by

$$IL = 20 \log \left(\frac{2K_1}{K_2}\right) \frac{1}{1 - \left(\frac{\omega}{\omega_0}\right)^2} , \qquad (1)$$

where K_2 and K_1 are the stiffness of the two-stage and single-stage mount isolators, respectively, and ω_0 is the resonant frequency of the blocking mass on the isolators. This expression applies only if the engine and frame rail mounting points are rigid. The insertion loss, calculated using this expression, is illustrated schematically in Fig. 38 under the assumption that the same isolators were used in both single- and two-stage mounts. Around the resonant frequency ω_0, the two-stage mount actually transmits more vibration than a single-stage mount. Above ω_0 the insertion loss increases rapidly. Accordingly, one usually seeks to make ω_0 as low as possible. In practice, the isolator stiffness cannot be made too small, because the engine

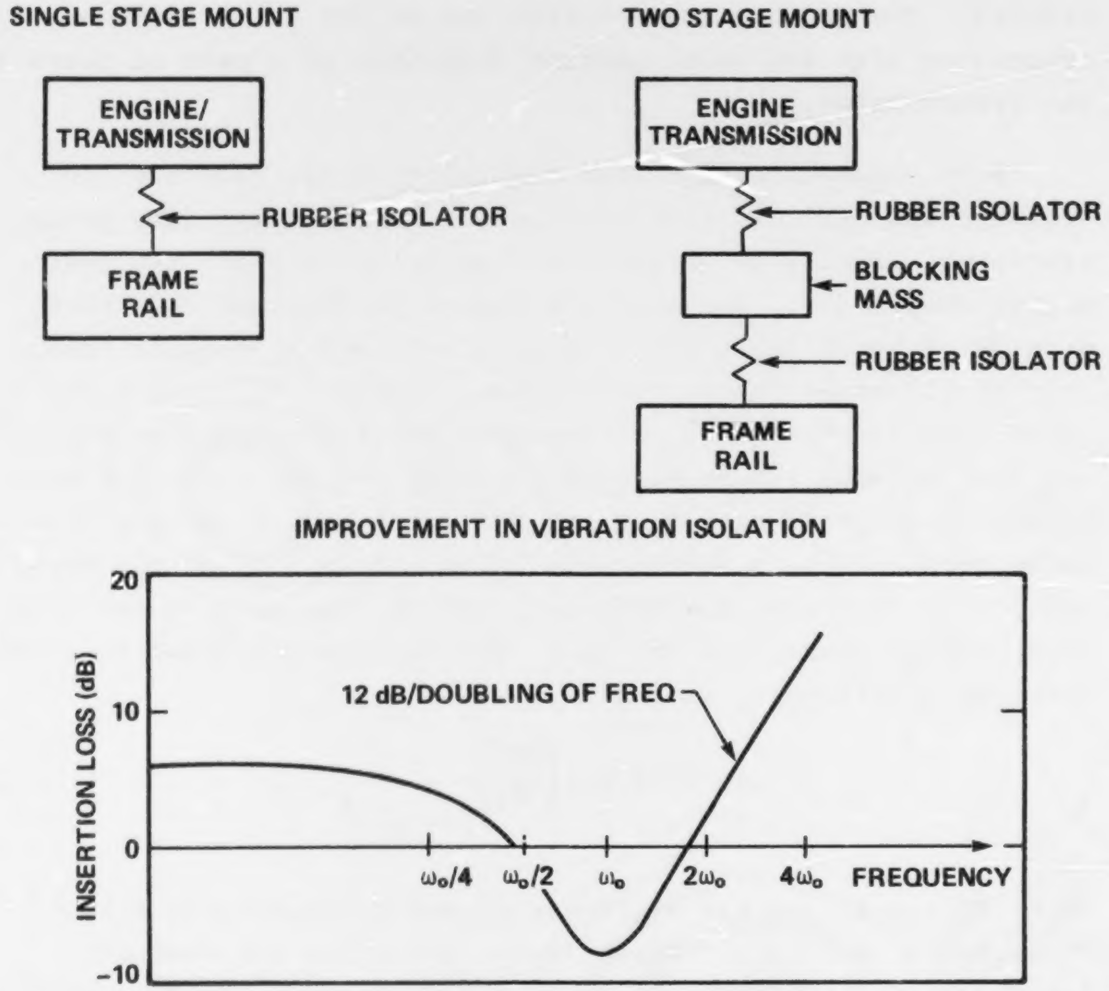

SINGLE STAGE MOUNT
ENGINE/ TRANSMISSION
RUBBER ISOLATOR
FRAME RAIL
TWO STAGE MOUNT
ENGINE TRANSMISSION
RUBBER ISOLATOR
BLOCKING MASS
RUBBER ISOLATOR
FRAME RAIL
IMPROVEMENT IN VIBRATION ISOLATION
INSERTION LOSS (dB)
20
10
0
-10
12 dB/DOUBLING OF FREQ
ω₀/4
ω₀/2
ω₀
2ω₀
4ω₀
FREQUENCY

FIG. 38. SCHEMATIC ILLUSTRATION OF SINGLE- AND TWO-STAGE ENGINE MOUNTS.

mounts must be stiff enough to support the loaded engine within its clearance envelope. Similarly, the mass cannot be made too large, because of weight and space restrictions.

The design objectives for the two-stage engine mounts were:

- Adequate reduction of truck frame vibration due to engine/transmission excitation

- Adequate restraint of the engine during peak torque operation and dynamic excitation from the roadway
- Durability
- Simplicity
- Minimum weight penalty.

In developing the two-stage engine mount, we retained the original transmission bracket (see Sec. 2.1) and designed new components for the remainder of the assembly. Figures 39 and 40 show the new configuration for the mount. The transmission

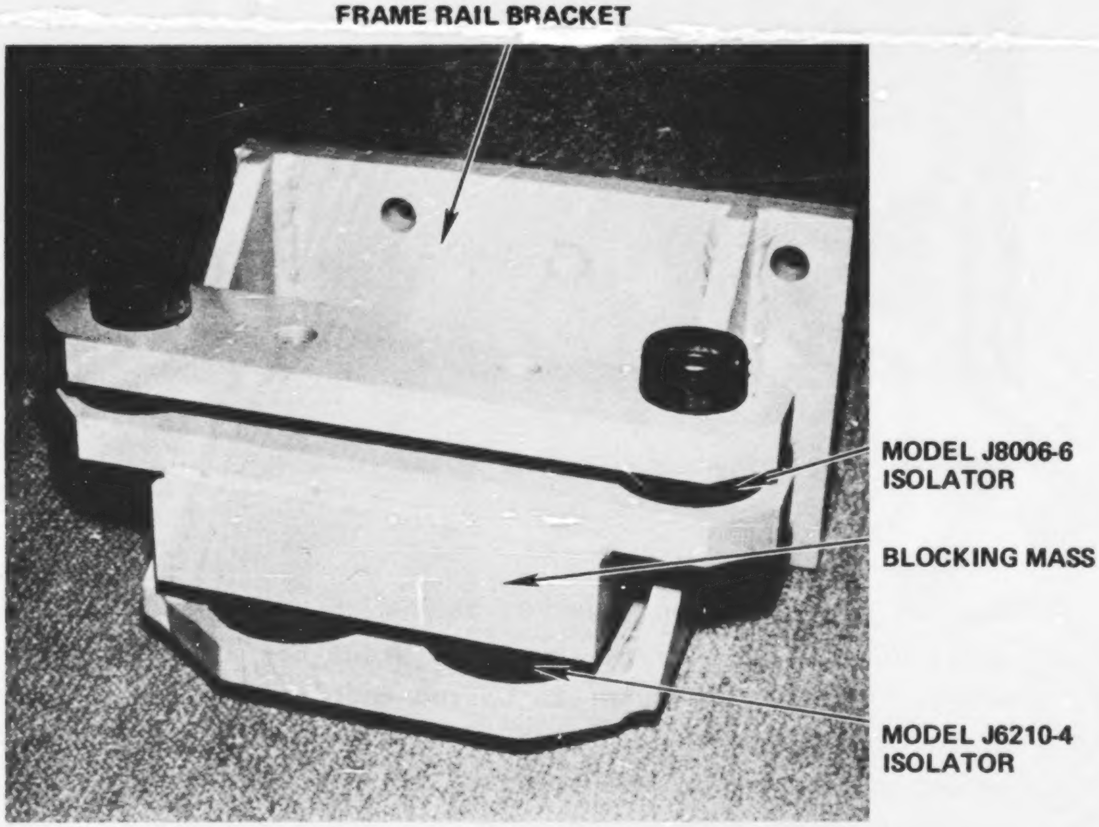

FRAME RAIL BRACKET
MODEL J8006-6 ISOLATOR
BLOCKING MASS
MODEL J6210-4 ISOLATOR

FIG. 39. THE ASSEMBLED TWO-STAGE ENGINE MOUNT.

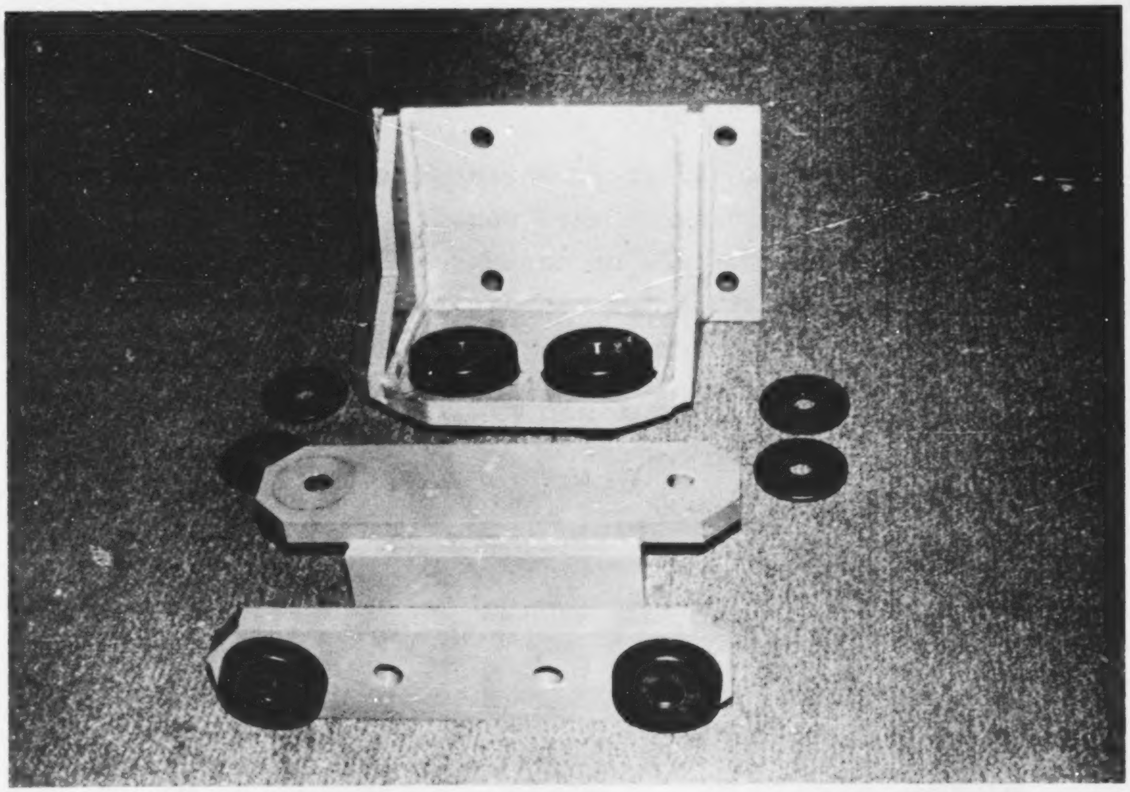


FIG. 40. THE TWO-STAGE ENGINE MOUNT DISASSEMBLED.

bracket (not shown in Figs. 39 or 40) is bolted to the threaded holes visible in the 3/4-in. thick steel bar at the top of the mount. The bar, in turn, is bolted through two rubber isolators (Lord Corporation Model J8006-6) at each end to a comparable bar welded to the top of the steel blocking mass. The bar and mass together weigh approximately 20 lb. The specially fabricated frame rail bracket at the bottom of the assembly is also bolted to the blocking mass through a pair of rubber isolators (Lord Corporation Model J6210-4).

Figure 41 shows the one-third octave band spectrum of the acceleration on the frame rail aft of the engine mounts before and after installation of the two-stage mounts. Although the reduction in vibration below 1000 Hz due to the two-stage mounts is modest, there is a 3-dB reduction at 500 Hz, the frequency that contributes most strongly to structureborne noise (see Appendix C). Differences of up to 10 dB are apparent at high frequencies.

The differences illustrated in Fig. 41 are less than one would expect from the theoretical considerations presented above. The reason probably relates to a flanking path through the forward engine mount, which originally contained a rubber isolator, but which was not treated further as part of this project. As discussed in Appendix C, measurement data suggest that the two-stage mounts provide less than 1 dBA of structure-borne noise reduction on the noisiest side of the truck. These mounts alone are probably not cost-effective. If further isolation were to be considered, it should include the front mount.

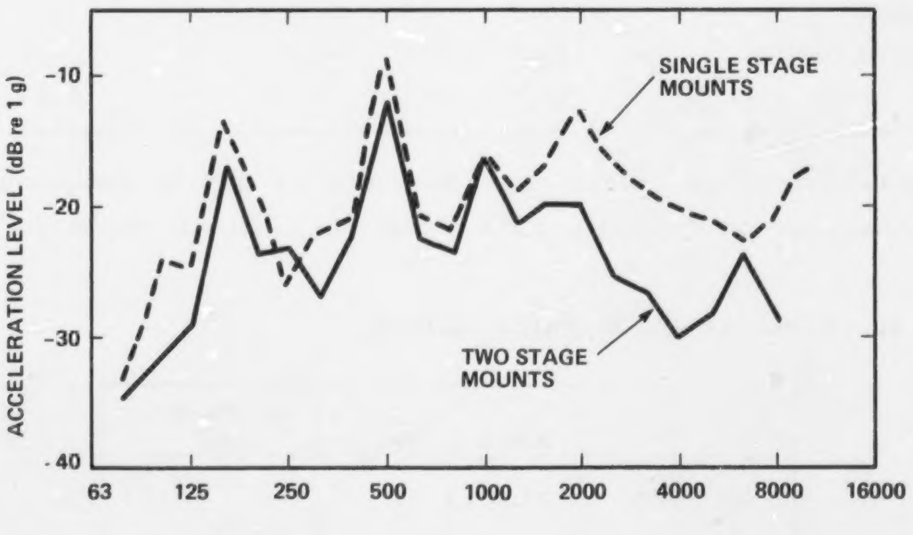


FIG. 41. CHANGE IN AFT FRAME RAIL VIBRATION DUE TO INSTALLATION OF TWO-STAGE ENGINE MOUNTS.

4. FINAL NOISE LEVELS

Measurements of exterior and interior noise levels were conducted according to the procedures described in Appendix A of this report. The results are reported here.

4.1 Exterior Noise Levels

Table 7 summarizes the noise source contributions for the initial and final vehicle configurations. The 8.4-dBA reduction in overall vehicle noise was achieved through an 8.8-dBA reduction in engine/transmission noise and a 9.5-dBA reduction in exhaust noise.

TABLE 7. SUMMARY OF NOISE SOURCE CONTRIBUTIONS.

Noise Source	Initial Level (dBA)	Final Level (dBA)	Noise Reduction (dBA)
Engine/Transmission	81.2	72.4	8.8
Exhaust	70.5	61.0	9.5
Intake	52.0	52.0	-
Other (coastby)	63.5	63.5	-
Total	81.6	73.2	8.4

Exterior noise levels were measured by BBN in Cambridge, Massachusetts, on July 31, 1981. The results are shown in Table 8.

TABLE 8. FINAL EXTERIOR NOISE LEVELS.

	Run 1	Run 2	40 CFR 205 Level
Left Side	73.1	73.2	73.2
Right Side	73.0	72.8	

4.2 Interior Noise Levels

Figure 42 shows the SAE J336a criteria [8] and the octave-band interior noise levels measured after the application of noise treatment. The criteria band levels shown in the figure are those that are summed to establish an overall criterion against which actual levels are to be compared. The maximum allowable band levels, established by the SAE J336a Recommended Practice, are not to be exceeded if the vehicle is to meet the design criteria.

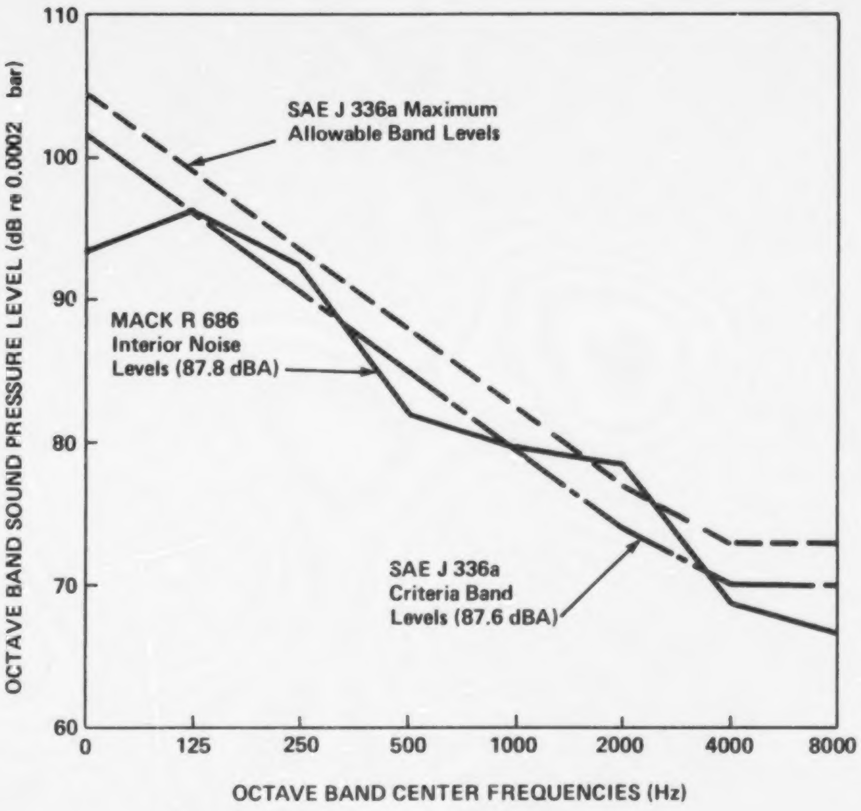


FIG. 42. INTERIOR NOISE LEVELS COMPARED WITH SAE J336a CRITERIA.

The data in Fig. 42 show that the interior levels are close to the criteria levels and that the overall 87.8-dBA level is approximately the same as the 87.6-dBA level corresponding to the criteria bands. However, the cab level exceeds the maximum allowable level in the 2000-Hz band.

5. PERFORMANCE FACTORS

The Mack R686 did not enter fleet service, because the original program was shortened. However, three factors relevant to its performance in a fleet were evaluated through laboratory and analytical studies and are reported in this section. These factors are (1) an estimate of the impact of noise treatments on fuel consumption, (2) an evaluation of engine mount load capacity, and (3) an assessment of vehicle serviceability.

5.1 Estimated Impact on Fuel Consumption

Several aspects of the noise control treatment may contribute to changes in vehicle fuel economy. The increased weight associated with the dual exhaust system and the engine/transmission enclosure adds to the rolling resistance, which, in turn, results in the need for a greater energy expenditure to haul a given load. The enclosure may either reduce or increase aerodynamic drag, which will similarly affect fuel consumption. The backpressure generated by the exhaust system will influence engine efficiency and associated fuel consumption. Here we estimate the magnitude of the effects of noise treatment on fuel consumption.

To estimate the additional fuel cost associated with additional weight, we consider the approximate relation between fuel consumption and weight presented in Ref. 13. By using a least-squares regression technique, Fax and Kaye fit a straight line to field data from a range of operations to derive the average fuel consumption sensitivity of

$$\Delta GPM/\Delta GCW = 1.45 \times 10^{-6} \text{ gal/mile/lb} ,$$

where ΔGPM is the incremental fuel consumption in gal/mile and ΔGCW is the incremental gross weight.

As will be shown in Sec. 6 of this report, the total weight increase associated with the noise treatment is 398 lb. Using this value in the above equation gives an expected change in fuel consumption of 5.77×10^{-4} gal/mile. At a nominal 5 mpg fuel consumption for an operating vehicle, this represents a 0.29% increase in fuel consumption.

To estimate the effect of backpressure, consider the relationships between fuel efficiency and backpressure illustrated in Fig. 43. The shaded area corresponds to a published composite of data [9], while the three curves within this area are for proprietary data supplied to BBN by several engine manufacturers. Reference 9 suggests that fuel economy improves by an average rate of 0.5% per inch of mercury (Hg) decrease in backpressure. This number, which is consistent with the data in Fig. 43, was used for our estimates.

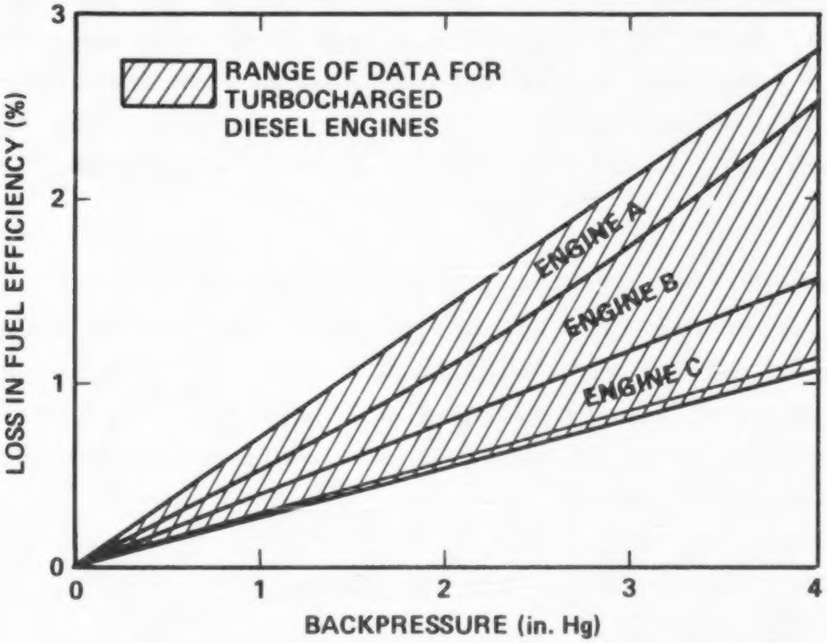


FIG. 43. RELATIONSHIP OF DIESEL ENGINE FUEL EFFICIENCY TO EXHAUST BACKPRESSURE.

The backpressure generated by the original exhaust system, measured under laboratory conditions on an ENDT 676 engine, was 0.75-in. Hg. Flow tests performed on the 5080B668 muffler and stack silencer show that they would generate 1.4-in. Hg at full load on an operating engine. When the difference of 0.65-in. Hg is multiplied by the 0.5%/in. Hg fuel penalty discussed above, the result is a 0.33% increase in fuel consumption.

Aerodynamic effects are not readily estimated on the basis of existing data. Wind tunnel tests of the vehicle or an accurate scale replica would be required to determine changes in drag, and such tests are beyond the scope of this program.

In summary, the estimated effects of noise control treatments are:

Estimated Increase in Fuel Consumption	
Weight	0.29%
Backpressure	0.33
Net	0.62%

5.2 Static Test of the Two-Stage Mount

To ensure the safe operation of the two-stage engine mounts during fleet service, we arranged with Teledyne Engineering Services, Waltham, MA, to perform a static load test on the frame rail bracket and lower isolators of one mount. Figure 44 shows the assembly located in a fixture in Teledyne's MTS electro-hydraulic test machine. During the test, the load was gradually increased. Figure 45 shows that at 25 kips the rubber isolators have undergone substantial deformation, but the bracket shows no visible displacement.

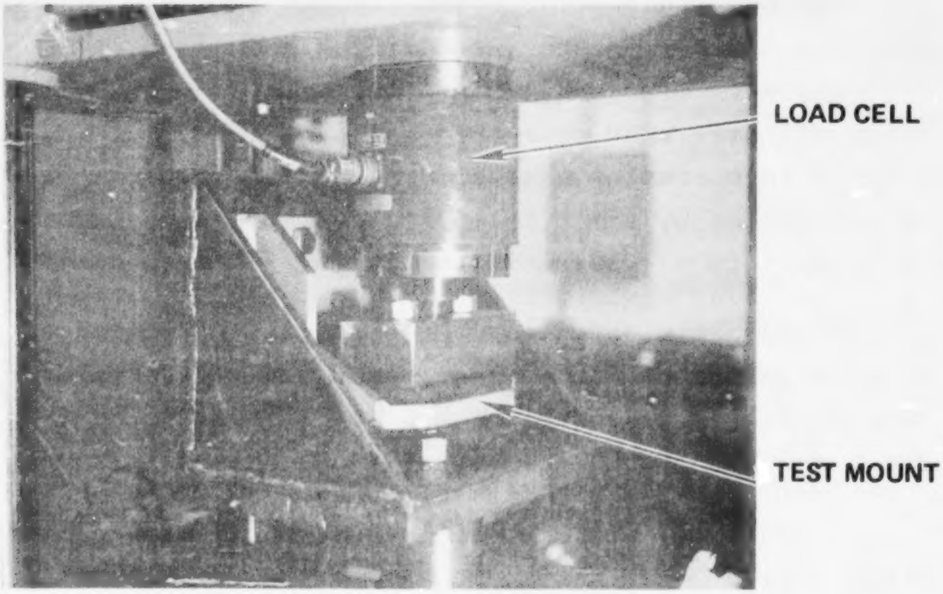

LOAD CELL
TEST MOUNT

FIG. 44. MOUNT ASSEMBLY IN LOAD TEST MACHINE.

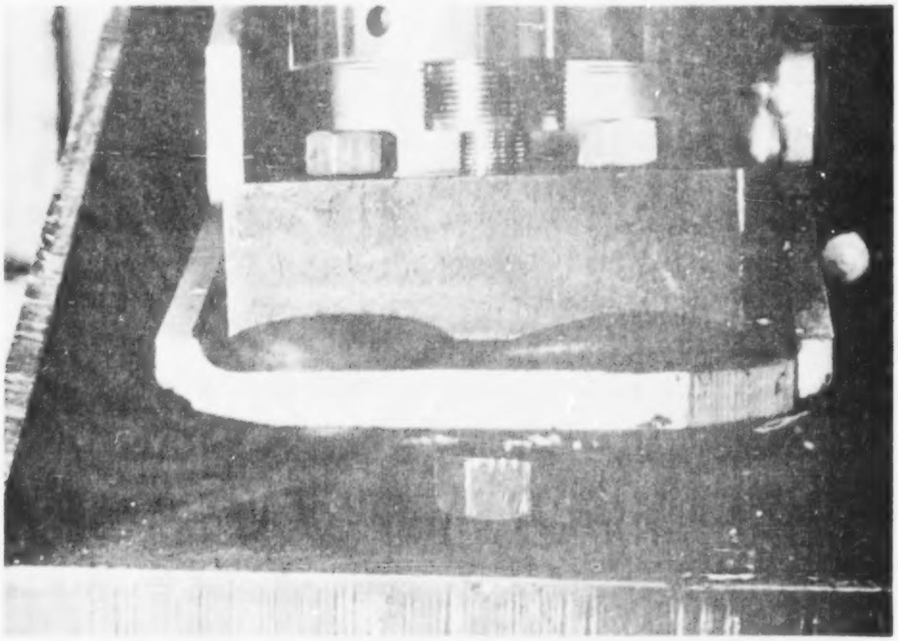


FIG. 45. TWO-STAGE MOUNT AND FIXTURES UNDER A STATIC LOAD OF 25 KIPS.

During the test, load and deflection were simultaneously recorded on an X-Y plotter, as illustrated in Fig. 46. A maximum load of 50 kips was reached, which is the maximum load capacity of the machine. Neither Fig. 46 nor a post-test visual inspection revealed any permanent deformation or damage to the mount.

Mack Trucks designs its mounts to accommodate peak torques with a safety factor of 2.0. During peak torque conditions the combined engine weight and torque is 5400 lb on the right isolator. Incorporating the factor of 2.0 results in a design limit of 10,800 lb, which our mount exceeds by a wide margin.

5.3 Serviceability

The serviceability of other vehicles in the Demonstration Truck Program was assessed in field tests during which the vehicles were placed in fleet service [10-12]. In this section, we present estimates of the impact of the treatments on the serviceability of the Mack R686, based on service frequencies for other trucks in the Demonstration Truck Program.

The enclosure is the only treatment that has an impact on serviceability. Neither the new exhaust system nor the modified engine mounts have an effect on serviceability. The enclosure affects maintenance in at least two ways:

- one or more bottom panels have to be removed to service the engine and transmission from below
- panels can restrict access of mechanics while performing normal maintenance.

To assess the serviceability impacts, we estimated the incremental time required to service the truck by conducting a time and motion study at a truck service facility in Boston. A

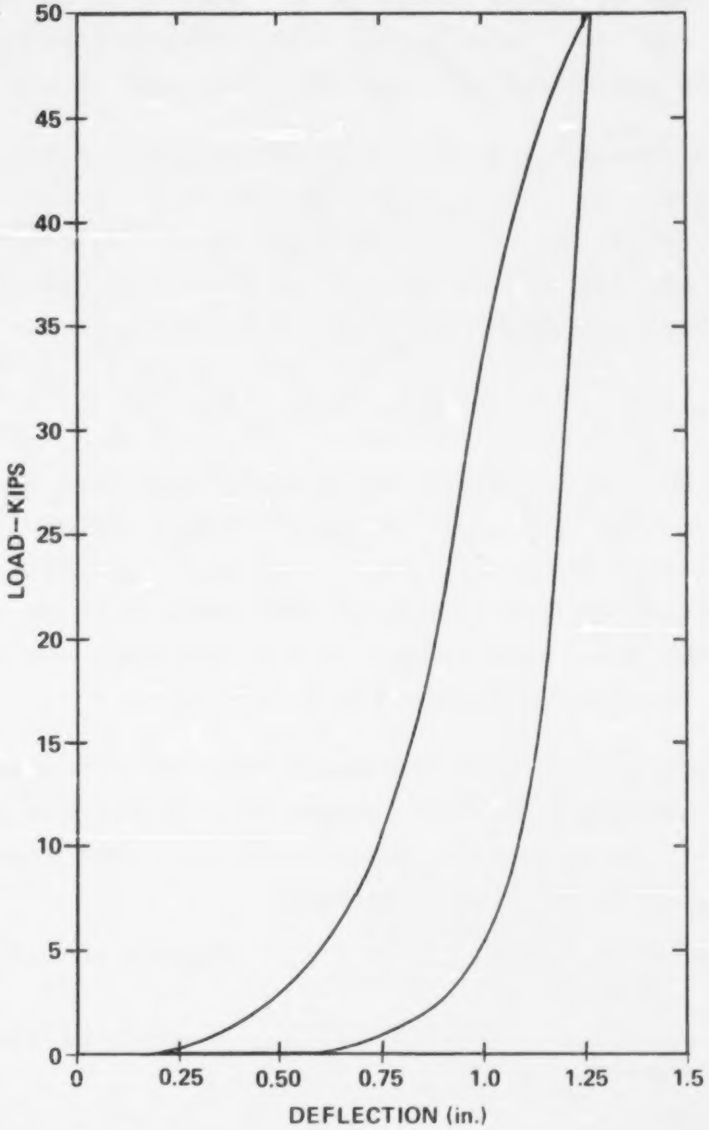


FIG. 46. LOAD/DEFLECTION CURVE FOR THE MACK TWO-STAGE MOUNT COMPONENTS ILLUSTRATED IN FIGS. 44 AND 45.

mechanic examined the enclosure and practiced taking the bottom panels off and replacing them. After he became familiar with the placement and mounting features of each panel, we timed him as he removed and reinstalled the panels. The results are presented in Table 9.

TABLE 9. TIME REQUIRED TO REMOVE AND INSTALL BOTTOM PANELS.

Panel*	Remove	Install
B1	0:19	1:10
B3	0:29	3:33
B4	0:59	4:34
B5	0:48	2:20
Total	2:35	11.37

*B2 is bolted in place and is not normally removed.

Bottom panels were typically removed once a month for trucks in the field-test phase of the Demonstration Truck Program. Given this assumed frequency of service and the times presented in Table 9, we estimate that the incremental time to remove and reinstall bottom panels would be 2 hours and 19 minutes per year. Restriction time penalties were 45% of removal and reinstallation time. Therefore, we estimate that the overall incremental time penalty for servicing the quieted Mack R686, i.e., both removal/reinstallation and access restrictions, would be 3 hours and 22 minutes.

Maintenance costs in the Demonstration Truck Program were charged at $17 to $17.75 per hour. Assuming a $17.50 rate, we estimate that the incremental service cost attributable to the noise control treatments would be $58.92 per year.

6. COST ESTIMATES

This section contains a discussion of the costs of the noise control treatments described in previous sections. There is a specific cost attributable to the manufacture and installation of each major noise control treatment: the engine/transmission enclosure, the two-stage engine mounts, and the modified exhaust system. We first present a summary of these costs, and then discuss the procedures used to estimate the cost of each treatment.

Table 10 presents the distinctions between costs and price used in this report. The convention is that the seller sells at a price, and a buyer buys at a cost. There are three sellers: the manufacturer of noise control products (e.g., a muffler manufacturer), the truck manufacturer, and the truck dealer. The three buyers are the the truck manufacturer, the truck dealer, and the truck operator. A markup is applied in moving from one level to another. Hence,

manufacturer's price × dealer markup = dealer's price.

TABLE 10. SUMMARY OF COSTS AND PRICES.

Transaction	Cost	Price
Sale of Component Supplier's Parts to Truck Manufacturer	Manufacturer Cost	Supplier Price
Sale of Truck by Manufacturer to Dealer	Dealer Cost	Manufacturer Price
Sale of Truck by Dealer to Operator/Customer	Operator Cost	Dealer Price

There is no single, generalized approach for cost estimation. The costing and pricing procedures of each truck manufacturer are highly confidential for competitive reasons.

Our approach to cost estimation is determined largely by the treatment to be costed and the availability of information from which cost estimates can be derived. All cost and price estimates are in 1979 dollars.*

6.1 Summary

Table 11 presents an overall summary of the treatment weights. Table 12 presents a summary of the estimated overall cost and price increases attributable to the noise control treatments installed on the Mack R686. The weight of the truck increased by 398 lb, approximately 2.5% of tractor tare weight (15,782 lb), or 0.5% of the 80,000-lb maximum permissible gross combination weight. The estimated price increase of $1,296 is a 3.2% increase over the actual purchase price of the vehicle, $40,757.

TABLE 11. SUMMARY OF TREATMENT WEIGHTS.

Treatment	Weight (lb)	Net Increase (lb)
Engine-Transmission Enclosure		244
• Components added	244	
Engine Mount Modifications		42
• Components added	42	
Exhaust System Modifications		112
• Components installed	189	
• Component removed	<77>	
Total weight	398	398

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*The vehicle is a 1979 model. Costs and prices are in 1979 dollars for consistency among the four technology and cost reports in this series.

TABLE 12. SUMMARY OF COST AND PRICE INCREASES.

	Net Increase	
Treatment	Dealer Cost ($)	Dealer Price ($)
Engine-Transmission Enclosure	630	946
Engine Mount Modifications	74	110
Exhaust System Modifications	177	240
Total	881	1296

The cost and price estimates presented here are BBN estimates for the retrofit treatments developed by BBN. They are not necessarily identical to the cost and price of treatments, were they to be installed by a truck manufacturer on production-level vehicles. There are several reasons why BBN cost estimates could differ from actual manufacturer costs. Each of the treatments is a tailor-made retrofit. More cost-effective design and materials specification by a manufacturer for actual production vehicles might well result in different treatment specifications and lower per-vehicle costs. While BBN has accounted for research, development, and testing (RD&T) and tooling costs by adjusting manufacturing cost estimates upward, that adjustment could be inaccurate, particularly if tooling or RD&T costs were atypical. The markup factors for manufacturers could differ among manufacturers from the markups assumed by BBN. Accordingly, the cost and price estimates presented here should be viewed as representative estimates for the treatments installed on the truck.

6.2 Enclosure Costs

Approach

The primary method of estimating the cost of the enclosure installed on the Mack R686 was to examine the relationship between the weight of materials and the cost of materials. This is a common technique used in engineering economics. Some components, such as special machined parts and electronic devices, have a price per pound greater than the overall price per pound of the truck; other components have a lower price per pound. Our focus is the weight-cost relationship for an enclosure. The first step is to obtain data with which to estimate a relationship. Having established a relationship, we then estimate the cost of the enclosure, given the weight of the enclosure.

We have presented elsewhere [3,4] a relationship between enclosure weight and manufacturer's price, with which one can estimate the cost of an enclosure. That relationship is a least-squares regression derived from data [13]. The estimated equation is:

$$Y = 61.3 + 1.92X \qquad R^2 = 0.99 \quad , \tag{2}$$

where Y is manufacturer's price in 1979 dollars and X is enclosure weight in pounds.

The coefficient of determination, designated R^2, can be interpreted as the variation in the dependent variable (manufacturer's price) accounted for by variation in the independent variable (enclosure weight). In this instance, 99% of manufacturer's price can be "explained" by enclosure weight. The estimated slope coefficient indicates that a 1-lb increase in weight would result in approximately a $1.92 increase in manufacturer's price (or a $2.88 increase in dealer price, given an assumed markup of 1.5 in going from manufacturer's price to dealer's price.)

This equation shows only the relationship between weight and manufacturer's price of a prototype enclosure. It does not include any costs for special tooling or research, development, and testing associated with commercial production of the enclosure.* Accordingly, any cost or price estimate derived from this equation is downward biased, since it excludes these costs. Conversely, it does not reflect any cost savings attributable to production economics.

Estimated Enclosure Costs

A summary of the components and weights for each assembly of the enclosure is presented in Table 13. The assembly weights presented in the table are based on either actual weight measurements by BBN or weight estimates derived from blueprint measurements and the weight of component material per unit area. As is evident from the table entries, the bulk of the weight increase is accounted for by fabricated aluminum components, which constitute the sides and bottom of the enclosure.

Given the enclosure weight of 244 lb and the weight-manufacturer's price relationship presented above, the estimated manufacturer's price of the enclosure is $530. This estimate is then increased by 19% to account for tooling and RD&T costs. The 19% escalation applied here is the same percentage applied in earlier reports in this series [3,4,5]. While tooling and RD&T

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*These costs are estimated separately in the following section.

TABLE 13. SUMMARY OF ASSEMBLY AND COMPONENT WEIGHTS (LB) - MACK R686 ENCLOSURE.

Treatment Code	Assembly	Components	Component Weight	Assembly Weight
L1	Left side shield	0.125 Alum.	5.5	5.5
R2F	Forward right side shelf	0.125 Alum. Forms top seal	0.8 0.1	0.9
L2 R2A	Left and aft right side shelves (matched pair)	0.16 Alum. sheet (2 @ 3.2)	6.4	6.4
R3 & L3	Front side panels (matched pair)	0.16 Alum. panels (2 @ 13.0)	26.0	26.0
R4 & L4	Middle side panel (matched pair)	0.16 Alum. panels (2 @ 11.9)	23.8	23.8
R5	Right rear side panel	0.16 Alum. panel	19.8	19.8
L5	Left rear side panel	0.16 Alum. panel	11.1	11.1
R6	Under cab panel - right	0.16 Alum. panel	5.1	5.1
L6	Under cab panel - left	0.16 Alum. panel	4.8	4.8
R7	Absorptive treatment on right rear panel	0.16 Perf. Alum. sheet 2 in. Channel Spacer 2 in. Mylar-covered Fiberglas	1.6 1.2 1.3	4.1
L7	Absorptive treatment on left rear panel	0.16 Perf. Alum. sheet 2 in. Channel Spacer 2 in. Mylar-covered Fiberglas	1.8 1.2 1.3	4.3
L8	Absorptive treatment on left middle panel	0.16 Perf. Alum. 2 in. Channel Spacer 2 in. Mylar-covered Fiberglas	1.5 1.2 1.2	3.9

TABLE 13. SUMMARY OF ASSEMBLY AND COMPONENT WEIGHTS (LB) - MACK R686 ENCLOSURE (Cont.)

Treatment Code	Assembly	Components	Component Weight	Assembly Weight
B1	Front bottom panel	0.125 Alum. sheet	11.1	11.1
B2	Forward vertical bottom panel	0.125 Alum. sheet	5.2	5.2
B3	Oil sump bottom panel	0.125 Alum. sheet	9.8	9.8
B4	Middle bottom panel	0.125 Alum. sheet	20.2	20.2
B5	Rear bottom panel	0.125 Alum. sheet	11.5	11.5
B6	Aft vertical bottom panel	0.125 Alum. sheet	6.3	6.3
T1	Walk plate	0.19 Alum. trend	20.6	20.6
T2	Absorptive treatment on walk plate	0.16 Perf. Alum. sheet 2 in. Channel Space 2 in. Mylar-covered Fiberglas	2.0 1.2 1.4	4.6
-	Front center seal between radiator and bottom panel	0.20 Alum. sheet	7.9	7.9
-	Assorted mounting brackets	0.25 Alum. 0.31 Alum. 0.75 Alum. 0.25 Steel 0.375 Steel	5.2 1.1 3.9 3.3 6.1	5.2 1.1 3.9 3.3 6.1
-	Under-hood absorptive treatment	2 in. Fiberglas	8.8	8.8
	Misc. seals and gaskets	Sheet Rubber Foam Seals	1.5 1.0	2.5
Total Weight				243.8

costs are influenced by a variety of factors, such as the complexity of the enclosure design, the materials used, and the volume of production, the 1.19 markup used in prior reports in this series has been accepted by reviewers of those reports. A 1.5 markup is then applied to manufacturer's price to obtain dealer price, estimated to be $946. The calculations are summarized as follows:

61.3 + 1.92(244) = $520.78
× 1.19 tooling and RD&T markup
$730.44 manufacturer's price
×1.50 dealer markup
$945.66 dealer price (3)

6.3 Engine Mounts

BBN installed two-stage mounts on the Mack R686. As described in Sec. 3, the main material differences between the standard Mack mounts and the BBN mounts were a 13.8-lb mass, a 7.3-lb plate, and a new frame rail bracket, which were added to each standard rear mount. Large rubber isolators and longer mounting bolts are also part of the BBN two-stage mounts.

The two-stage mounts installed by BBN on the Mack R686 have the same conceptual design as two-stage mounts designed by BBN for other quieted trucks, specifically the International Harvester F-4370 in the EPA Demonstration Truck Program and the Freightliner truck in the DOT Quiet Truck Program. The major difference in the mounts among the three trucks is size and weight.

The cost of the Mack two-stage mounts is estimated using the same procedure used to estimate the cost for the IH F-4370 two-stage mounts [5]. The manufacturer's price per pound in 1979 dollars has been determined to be $1.47, an estimate based on results from the Freightliner and IH cost analyses. This estimate is then factored upward by 19% for RD&T costs and for dealer markup. These markups are the same as those for estimating the costs and prices of engine enclosures.

The incremental weight of the Mack two-stage mounts is 42 lb. Given this weight and the markups identified above, the price of mounts is estimated as follows:

42 lb × $1.47/lb =	$61.74	
	1.19	Tooling and RD&T markup
	73.47	Manufacturer's price
	x 1.50	Dealer markup
	$110.21	Dealer price.

6.4 Exhaust System Costs

The baseline exhaust system configuration of the R686, described in Sec. 2, included a single, vertical, aluminized muffler. BBN replaced that muffler and modified other exhaust system components to achieve a reduction in exhaust outlet noise of 12.4 dBA. In this section we estimate the cost and price of those modifications.

Table 14 presents a summary of the exhaust system components that were removed and installed and the weight of each component. The BBN modifications increased the overall weight of the exhaust system by 112 lb. The largest single component was a 94-lb muffler. This oversized muffler is approximately 30 lb heavier than the mufflers installed by BBN on other trucks in the Demonstration Truck Program, and 70 lb heavier than the original baseline muffler.

The other major weight gain was the modified exhaust system piping from the turbo to the flange at the base of the muffler.

As described in Sec. 3.1, BBN substantially modified the baseline system. These changes resulted in a net increase of approximately 47 lb.

A different procedure has been used to estimate the cost of the exhaust system treatments. Other trucks in the Demonstration Truck Program had been delivered with stock exhaust systems that were one of several exhaust system options available for each vehicle. BBN could estimate the cost and price of its system by comparing it to cost of other optional systems for each vehicle. In contrast, the Mack's baseline exhaust system was atypical, in that it was designed for this application. BBN did not install a dual system but rather modified the baseline system using

TABLE 14. SUMMARY OF EXHAUST SYSTEM COMPONENTS AND WEIGHTS.

Component	Weight (lb)
Installed	
Super Stack Silencer	10.0
Muffler & Extension to Flange	94.0
Modified Exhaust Pipe-Flange to Bulkhead (incl. covered stack silencer)	67.3
Flex Pipe - Bulkhead to Turbo	9.3
Clamps (8)	8.1
Gross Increase	188.7
Removed	
Tailpipe	< 10.4>
Muffler & Extension to Flange	< 24.3>
Heat Shield	< 12.0>
Exhaust Pipe - Flange to Turbo	< 30.0>
Gross Decrease	76.7
Net Increase	112.0

components that were either not generally available, or were specially fabricated for the BBN system.

Donaldson Corporation supplied to BBN confidential price information to be used for "computational purposes." BBN used this information to estimate the cost of components removed from the original exhaust system and installed in the modified system. Since several components were not standard items or had been specially fabricated, BBN estimated their cost by comparing their specifications to standard items and using the cost of the standard items. For example, the muffler installed by BBN is not a standard truck muffler, but it is comparable to Donaldson muffler model WOM12-0284. The cost of other system components was estimated in similar fashion. These costs were at the supplier's price level - i.e., the price at which an exhaust system supplier would sell to a truck manufacturer. A manufacturer's markup of 40% and a dealer's markup of 35% are then applied. These markups are based on examination of price data supplied by Donaldson and published cost and price lists for exhaust system options for several truck manufacturers.

Applying this procedure to the components in Table 12, we estimate the net overall increase in dealer price of the exhaust system modifications to be $240.00. The additional components have an estimated dealer price of $352; this is offset by $112 of components that were removed.

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APPENDIX A: TEST REQUIREMENTS

Two procedures have been followed in testing the truck for noise. Exterior noise is measured according to the procedure described in 40 CFR 205, which is very similar to the SAE J366b Recommended Practice. Interior noise is measured according to the SAE J336a Recommended Practice. These test procedures are described in considerable detail in documents which should be consulted by readers who wish to understand them fully (see Refs. 7 and 8 of main report). Here we describe the major features of each test.

A.1 Exterior Test (40 CFR 205)

The exterior test is a low-speed, full-throttle acceleration test intended to characterize drive train noise while deemphasizing tire and aerodynamic noise. The general arrangement of the test site is illustrated in Fig. A.1. The site is comprised of a paved vehicle path and measurement area, surrounded by an area that is free of reflecting objects. A microphone is located 4 ft above the ground and 50 ft from the center of the vehicle path. During a test, the vehicle is driven along a straight path at a constant speed corresponding to approximately two-thirds of governed engine speed. At the Acceleration Point the throttle is opened fully. The vehicle accelerates through the next 100 ft, reaching maximum governed rpm in the test zone. The truck is operated in the highest gear step that will permit it to meet this requirement. The peak noise level is generally measured twice on each side, and the highest of the average values for each side is reported. Precision sound measuring equipment is used to ensure that accurate data are acquired.

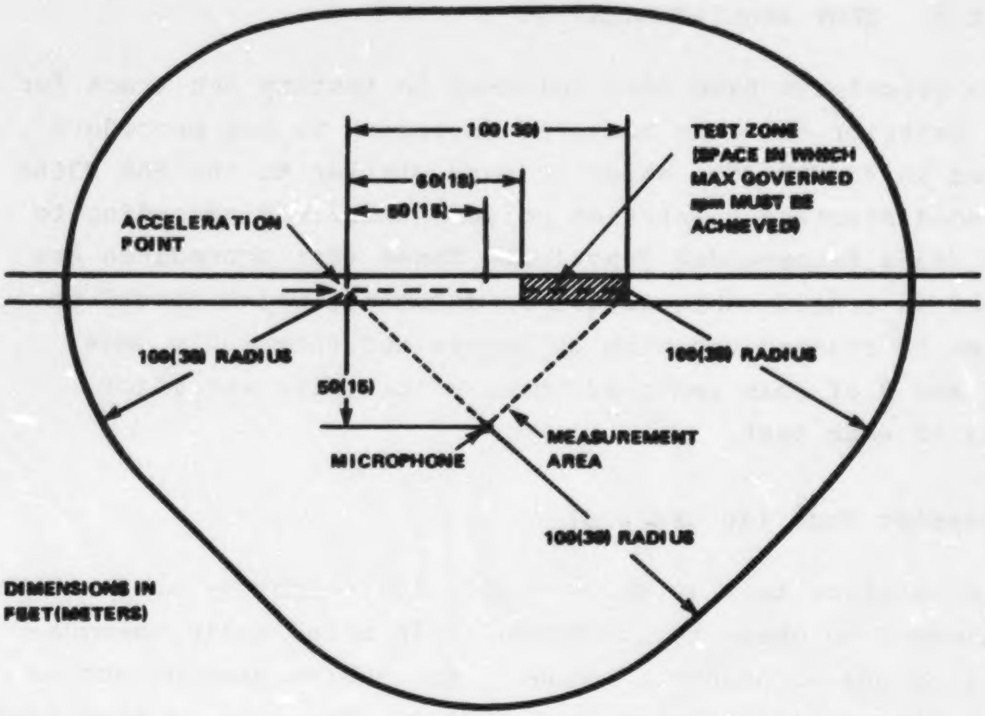
100(30)
60(18)
50(15)
TEST ZONE (SPACE IN WHICH MAX GOVERNED rpm MUST BE ACHIEVED)
ACCELERATION POINT
100(30) RADIUS
100(30) RADIUS
50(15)
MICROPHONE
MEASUREMENT AREA
100(30) RADIUS
DIMENSIONS IN FEET(METERS)

FIG. A.1. TEST SITE FOR EXTERIOR NOISE LEVEL MEASUREMENTS.

For the noise data reported here, the following operating conditions apply:

Engine Speed	- approach:	1100*rpm
	- final:	2250 rpm
Vehicle Speed	- approach:	10 mph
	- final:	21 mph
Gear Step:		3rd*

*The gear step and approach engine speed were determined experimentally as required by the test procedure. It was found that when the truck approached in fourth gear, with the engine running two-thirds of governed speed, the engine reached governed speed when the vehicle was beyond the test zone. In second gear and two-thirds of governed speed, the engine reached governed speed before the test zone. Accordingly, the engine speed at approach was successively reduced in 100-rpm increments until it was found that, at 1100 rpm, governed speed was reached within the test zone.

An important feature of this test procedure is that it allows thermostatically controlled radiator fans to remain inoperative. Accordingly, the fan clutch hub was disengaged, permitting the fan to turn only at a low speed, at which its noise contribution was judged inconsequential.

A.2 Interior Test (SAE J336a)

The SAE J336a Recommended Practice specifies noise measurements 6 in. from the driver's ear, while the truck is accelerating at full throttle, from approximately 25 mph to 50 mph [8]. The gear step is selected so that the engine reaches rated speed at 50 mph. The test is performed with windows and vents closed and accessories turned off. Because of the relatively high speed at which the test is conducted, one may expect tire noise to be a more significant part of the total measured level than in the case of the 40 CFR 205 or SAE J366b test procedures.

The SAE J336a test procedure does not require the reporting of the A-weighted level, but rather the average of the two highest levels in each octave frequency band. The following table illustrates the band center frequencies for which measurements are to be acquired and the band pressure levels to be considered during the development of new vehicles.

Octave Band Center Frequency (Hz)	Band Pressure Level (dB)	Octave Band Center Frequency (Hz)	Band Pressure Level (dB)
63	101.5	1000	79.5
125	96.0	2000	74.0
250	90.5	4000	70.0
500	85.0	8000	70.0

The Recommended Practice states that "Trucks meet the design criteria if the sum of reported band pressure levels does not exceed the sum of the criteria band pressure levels, provided that no reported band pressure level exceeds the corresponding criteria band level by more than 3 dB." While the Recommended Practice does not specify an A-weighted criterion, the (logarithmic) sum of the A-weighted values of the band pressure levels specified in the above table is 87.6 dBA.

APPENDIX B: ESTIMATION OF SOURCE CONTRIBUTIONS IN THE MACK R686.

In this appendix, we describe how the contributions from the noise sources on the Mack R686 were estimated from the various field measurements that were carried out on the truck. The estimates here are for source contributions when the truck is operated according to the 40 CFR 205 test procedure. Table B.1 presents a description of each source and the variables that will be used to represent each.

TABLE B.1. SOURCES ON THE MACK R686.

Variable	Source Description
EXO	Exhaust outlet noise
EXS	Exhaust noise radiated from the exhaust pipe shell
I	Engine intake noise
ICB	Coastby noise, i.e., tires, differentials, and airflow
ENB	Airborne noise coming from the back opening of the enclosure
ENF	Airborne noise coming from the front opening of the enclosure
ENR	Residual airborne noise escaping from the enclosure after the front and rear are sealed
SB_i	Structureborne noise from the engine and transmission radiated by the truck structure excluding the enclosure, i = 1 for single-stage mounts; i = 2 for two-stage mounts
SBE	Structureborne noise from the engine and transmission radiated by the engine enclosure

After installation of the noise control treatments, the total noise from the truck is given by*

$$N = EXO \oplus EXS \oplus I \oplus CB \oplus ENF \oplus ENB \oplus ENR \oplus SBE \oplus SB \quad , \qquad (B.1)$$

where it has been assumed that intake and coastby noise are unaffected by the improved exhaust system, engine enclosure, and two-stage rear engine mounts that constitute the noise control treatments.

Airborne and Structureborne Noise from the Enclosure

To determine the noise contribution from the engine and transmission, we separated the airborne contributions into three parts and the structureborne contributions into two parts. Engine/transmission airborne noise reaches the microphone through the opening in the front of the enclosure ENF; through the opening in the back of the enclosure ENB; and through leaks between the enclosure panels and other openings in the enclosure ENR. Structureborne noise is radiated by the enclosure panels, SBE, and other components of the truck structure, such as frame rails, cab, and fuel tanks.

With the truck in its final treated configuration except for the exhaust pipe, which was wrapped with leaded vinyl and fiberglass, a series of measurements was carried out with the front and rear of the enclosure alternately sealed with fiberglass and leaded vinyl and unsealed. Table B.2 presents the average noise levels from those tests in which the truck was operated according to the 40 CFR 205 test procedure. Also shown are the number of

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*The symbol $\oplus$ refers to logarithmic addition defined by

$$A \oplus B = 10 \log \left[10^{A/10} + 10^{B/10} \right]$$

runs from which the average noise levels were calculated, and the standard deviation, which provides an indication of the variability in the measurements. Because of the variability, we decided to take the average of a large number of runs rather than just the average of the first two, as called for in the EPA test procedure. If we assume that sealing the front and rear openings totally eliminates the noise from those paths, we can readily calculate ENF and ENB from the data in Table B.3 in two different ways. For example, let us define L_{Li} as the noise level on the left side of the truck with the truck in configuration 1 of Table B.2. Then, using one approach, which we will refer to as source suppression because we identify the source contribution by suppressing it relative to the other sources, we can write

TABLE B.2. NOISE LEVELS FROM THE MACK R686 WITH FRONT AND REAR OF ENCLOSURE SEALED AND UNSEALED.

		Left Side			Right Side		
Config. No.	Configuration Description	Noise Level (dBA)	Std. Dev. (dB)	No. of Runs	Noise Level (dBA)	Std. Dev. (dB)	No. of Runs
1	Front and rear sealed	73.6	0.2	6	73.4	0.5	4
2	Front open; rear sealed	74.1	0.2	5	73.7	0.4	4
3	Front sealed; rear open	73.7	0.4	5	73.4	0.2	5
4	Front and rear open	74.9	0.1	4	74.2	0.1	4

$$\begin{aligned}(\mathrm{ENF})_L &= L_{L4} \ominus L_{L3}\\ (\mathrm{ENB})_L &= L_{L4} \ominus L_{L2}\end{aligned} \tag{B.2}$$

Another approach, which we will call source enhancement because now we are enhancing the source of interest relative to the other sources, leads to the following equations:

$$\begin{aligned}(\mathrm{ENF})_L &= L_{L2} \ominus L_{L1}\\ (\mathrm{ENB})_L &= L_{L3} \ominus L_{L1}\end{aligned} \tag{B.3}$$

Those same equations can be applied to the calculation of ENF and ENB on the right side of the truck. Table B.3 presents the estimates of ENF and ENB using the above two equations for both sides of the truck. Since the two estimation procedures can result in different source strength values, the table also shows the best estimate of the source contributions. In the case of ENF, the estimate is simply the average of the two, since the two estimation procedures result in nearly the same source contributions. For ENB, opening the back with the front closed resulted in too small a change in noise to provide a reliable estimate of the noise from the back of the enclosure. Consequently, we have simply used the estimate based on Eq. (B.2). The

TABLE B.3. NOISE CONTRIBUTIONS FROM FRONT AND REAR OF ENCLOSURE.

Source	Left Side			Right Side		
	Source Contribution Eq.B.2	Source Contribution Eq.B.3	Estimated Source Contribution	Source Contribution Eq.B.2	Source Contribution Eq.B.3	Estimated Source Contribution
ENF	68.7	64.5	67.1	66.5	61.9	64.8
ENB	67.2	-	67.2	64.6	-	64.6

widely differing estimates of ENB that result from opening t back with the front closed and closing the back with the fro open are probably due to the variability of the noise from th particular truck.

We attempted to estimate the residual airborne noise from the enclosure ENR combined with the structureborne noise from th enclosure SBE, using measurements of the noise from the truck with the enclosure alternately wrapped with leaded vinyl and fiberglass and unwrapped. Table B.4 shows the average truck noise for that series of tests. The noise either did not decrease after the enclosure was wrapped or increased slightly. The increase in noise is probably a consequence of the variability in noise from the truck. On the basis of those data, we have assumed that

$$\text{ENR} \oplus \text{SBE} = \text{negligible noise contribution} \quad .$$

TABLE B.4. NOISE FROM TRUCK WITH ENCLOSURE WRAPPED AND UNWRAPPED.

	Left Side			Right Side		
Configuration Description	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs
Enclosure wrapped	72.9	0.3	6	73.4	0.2	4
Enclosure unwrapped	72.9	0.5	6	73.0	0.3	5

Exhaust Noise

Exhaust outlet noise is discussed in Sec. 3.1 of the test and determined to be 58.1 dBA. Exhaust shell noise, discussed in Sec. 3.2, is believed to have been reduced to a negligible level. Two tests, the results of which are shown in Table B.5, support this conclusion. In the first, the exhaust line, containing tubing and a 5-in. stack silencer, was completely wrapped with Fiberglas and leaded vinyl. This treatment is expected to reduce shell noise to a negligible level. In the second, Fiberglas was placed only around the stack silencer and was covered with a 6-in. pipe. The noise level for this final treatment was very nearly the same as for the leaded vinyl and fiberglass wrapping, demonstrating a comparable level of effectiveness in reducing shell noise to a negligible value.

TABLE B.5. NOISE FROM TRUCK WITH EXHAUST LINE WRAPPED AND UNWRAPPED.

	Left Side			Right Side		
Configuration Description	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs	Noise Level (dBA)	Standard Deviation (dB)	No. of Runs
Exhaust line wrapped	73.4	0.5	5	72.8	0.2	6
Exhaust line with final treatment	73.2	0.3	5	72.8	0.3	5

Structureborne Noise from the Truck

To estimate the structureborne noise from the truck structure exclusive of the enclosure, we have simply subtracted all the known sources estimated as described above from the overall noise from the truck, i.e.,

$$SB = N \ominus [EXO \oplus I' \oplus CB' \oplus ENF \oplus ENB] \ , \qquad (B.4)$$

where exhaust shell noise from the exhaust pipe, EXS; the residual airborne noise from the enclosure, ENR; and the structureborne noise from the enclosure have all been assumed to be negligible. The results of carrying out the above calculations for the right and left side of the truck are shown in Table B.6.

The results in Table B.6 for the left side of the treated truck are reasonably consistent, showing a slight decrease in structureborne noise from 69.3 dBA to 68.7 dBA. The results for the right side of the treated truck are not consistent, showing an increase in structureborne noise from 66.6 dBA to 70.0 dBA. We do not believe that such an increase actually occurred. Instead, our estimation procedure was probably affected by the variability in the noise that the truck makes from run to run and from day to day. Consequently, the high estimated structureborne noise on the right side of the truck is mainly due to the underestimation of the contribution from other sources.

TABLE B.6. SUMMARY OF SOURCE STRENGTHS OF TREATED MACK R686.

Source	Variable	Left Side Source Contributions (dBA)	Right Side Source Contributions (dBA)
Exhaust outlet	EXO	58.1	58.1
Exhaust shell	EXS	Negligible	Negligible
Intake	I'	52	52
Coastby	CB'	63.5	63.5
Airborne noise from the back of the enclosure	ENB	67.2	64.8
Airborne noise from the front of the enclosure	ENF	67.1	64.6
Residual airborne noise and structureborne noise from the enclosure	ENR ⊕ SBE	Negligible	Negligible
Structureborne noise from the truck	SB	68.7	70.0
Total		73.2	72.8*

*This value represents the average of 5 runs which we use for statistical accuracy, while the value of 72.9 reported in Table 8 represents an average of the first two runs as required by the EPA test procedures.

APPENDIX C: STRUCTUREBORNE NOISE FROM THE MACK R686.

Early in this program we realized that structureborne noise was an important source in the Mack. Consequently, we devised a series of tests to quantify the contribution of structureborne noise and to determine from which parts of the truck structure the noise was being radiated. A number of approaches were considered for obtaining the information. We first performed a vibration survey on the truck and used that information to estimate the sound radiation from the various components of the truck structure. It was determined that structureborne noise was likely to be a significant source but that accurate estimates of its level were required. We considered cover-and-expose measurements as well as direct measurements of sound intensity, e.g., acoustic-intensity or surface-intensity measurements. We decided to use the cover-and-expose approach. Although the direct measurement of sound intensity is emerging as a powerful diagnostic technique, its application to a group of noise sources that are moving and varying in intensity (as the sources on a truck are during the EPA test procedure) involves an extension of the state of the art. In addition, once the sound power radiated by each element of the truck structure has been measured, estimating the sound pressure at the 50-ft microphone location involves considerable uncertainty, because of complicated propagation and shielding effects.

On the other hand, the cover-and-expose approach, while somewhat cumbersome, does provide a well-established direct measure of the sound pressure at the 50-ft microphone location from sound radiated by each component of the truck structure. In this appendix, we first describe the preliminary estimates of the sound radiated by each component of the truck structure. In this appendix, we first describe the preliminary estimates of the structureborne noise from the truck, using the survey of vibration levels on the various components of the truck structure. We then describe the cover-and-expose measurements that more accurately quantified the structureborne contribution.

Preliminary Estimation of the Structureborne Noise

To make a preliminary assessment of the strength of the structureborne noise source contribution, we performed a vibration survey on the truck measuring the one-third octave band acceleration spectrum, $AL(\omega)$, at six locations while operating the truck according to the EPA test procedure. The locations measured were as follows:

- Bumper
- Fuel tank
- Cab
- Frame rail (two positions)
- Enclosure (Masonite).

The measured spectra are shown in Figs. C.1 and C.2.

The bumper acceleration was measured on the right side with the accelerometer oriented in the horizontal direction parallel to the axis of the truck, and the fuel tank acceleration was measured on the side of the right tank in a direction perpendicular to the truck axis. The cab vibration was obtained from measurements on the right door. Vibration levels on the fiberglass hood were generally 30 dB or more below one "g" in all one-third octave bands; therefore, radiation from the hood was not included in this calculation. Two positions on the frame rail were measured: one just aft of the bumper and one just forward of the rear tandem axle. For both, the accelerometers were

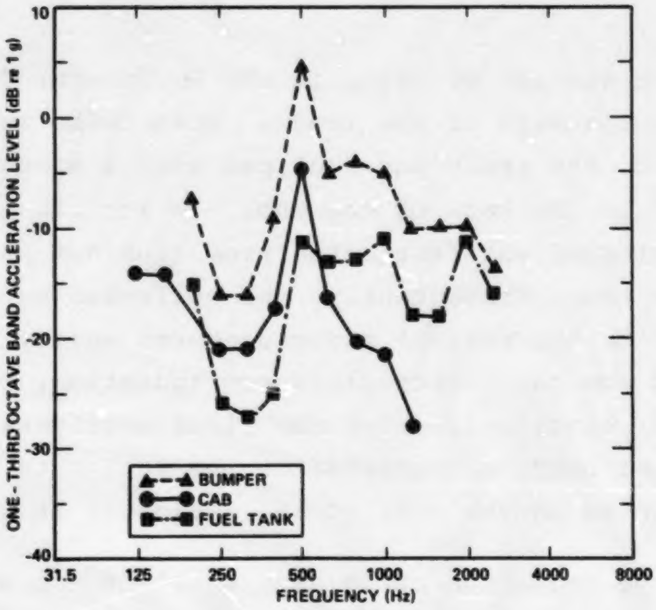

FIG. C.1. VIBRATION LEVELS ON CAB FUEL TANK AND BUMPER.

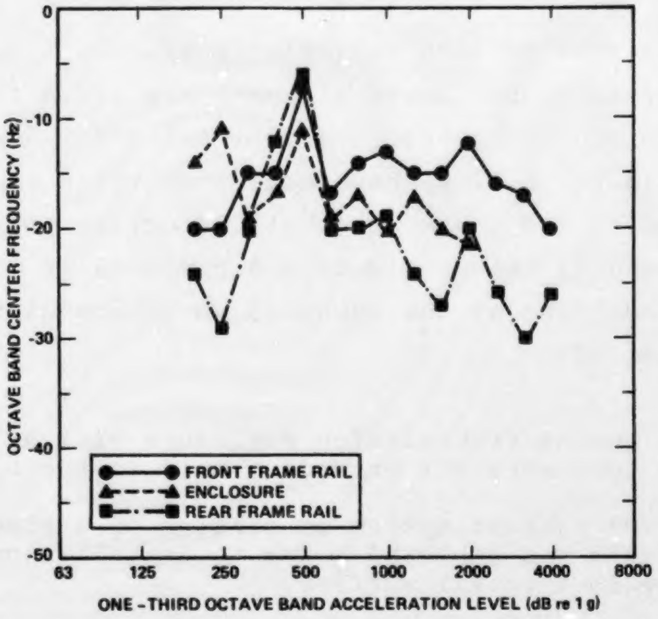

FIG. C.2. VIBRATION LEVELS ON FRAME RAIL AND ENCLOSURE.

located on the web and oriented in the horizontal direction perpendicular to the axis of the truck. When these measurements were performed, the truck was equipped with a Masonite enclosure that extended to the back of the cab. In its final configuration, the enclosure was fabricated from aluminum and extended 3 ft aft of the cab. Consequently, the estimated contribution of the enclosure to the overall structureborne noise, shown here, while correct for the intermediate configuration, probably underestimates that contribution for the final configuration. The vibration level used to characterize the enclosure sound radiation was measured on the side of the enclosure at the front axle.

Taking the radiation efficiency of all these surfaces as unity, we can estimate the sound pressure level, SPL(ω), at frequency ω at 50 ft to be

$$SPL(\omega) = 124 + AL(\omega) + 10 \log A - 20 \log \omega \quad , \qquad (C.1)$$

where A is the area of each radiating surface in square feet. The surface areas of the above elements are given in Table C.1. Using the acceleration levels in Figs. C.1 and C.2 and the areas in Table C.1 in Eq. C.1, we have estimated the structureborne noise radiated by the truck in one-third octave frequency bands. Figure C.3 presents that estimate and compares it to measurements of the noise radiated by the truck in an intermediate configuration consisting of:

- Masonite engine/transmission enclosure similar to the final aluminum enclosure but extending only to the back of the cab
- An improved exhaust system consisting of a single 10-in. muffler that was employed prior to installation of the final 10-in. by 15-in. oval muffler
- The original equipment single-stage engine mounts.

As the figure shows, structureborne noise appears to be a significant contributor to truck noise up to 800 Hz, with a significant peak occurring in the 500-Hz band. We believe that the transmission is responsible for the 500-Hz peak. During the tests performed according to the EPA test procedure, the truck is operated in third gear. At rated engine speed (2100 rpm), the tooth passage frequency of the three countershaft gears meshing with the output shaft gear is 518 Hz, which strongly implicates the transmission as the source of excitation in the 500-Hz band.

TABLE C.1. SURFACE AREA OF TRUCK COMPONENTS [FT^2].

Bumper	14.5
Fuel Tanks	34.0
Battery Boxes	15.7
Cab (excluding doors)	21
Frame Rail (vertical)	23.7
Frame Rail (horizontal)	39.5

The contribution of the five radiating surfaces to the overall structureborne noise is shown in Table C.2. The estimates indicate that the frame rail, cab, and bumper are the primary radiators.

TABLE C.2. CONTRIBUTION OF VARIOUS RADIATING SURFACES TO STRUCTUREBORNE NOISE.

Source	Contribution (dBA)
Bumper	70.2
Cab	67.3
Frame Rail	66.5
Fuel Tanks	62.2
Enclosure	61.3
Total	73.7

As Fig. C.3 shows, the preliminary estimates indicate significant structureborne noise below 800 Hz and a very strong contribution in the 500-Hz one-third octave band. The estimated structureborne contribution of 73.7 dBA is unacceptably high if we are to reduce overall truck noise to 72 dBA. However, this preliminary estimate is subject to some uncertainty and is in fact an upper-bound estimate, since we have made no effort to

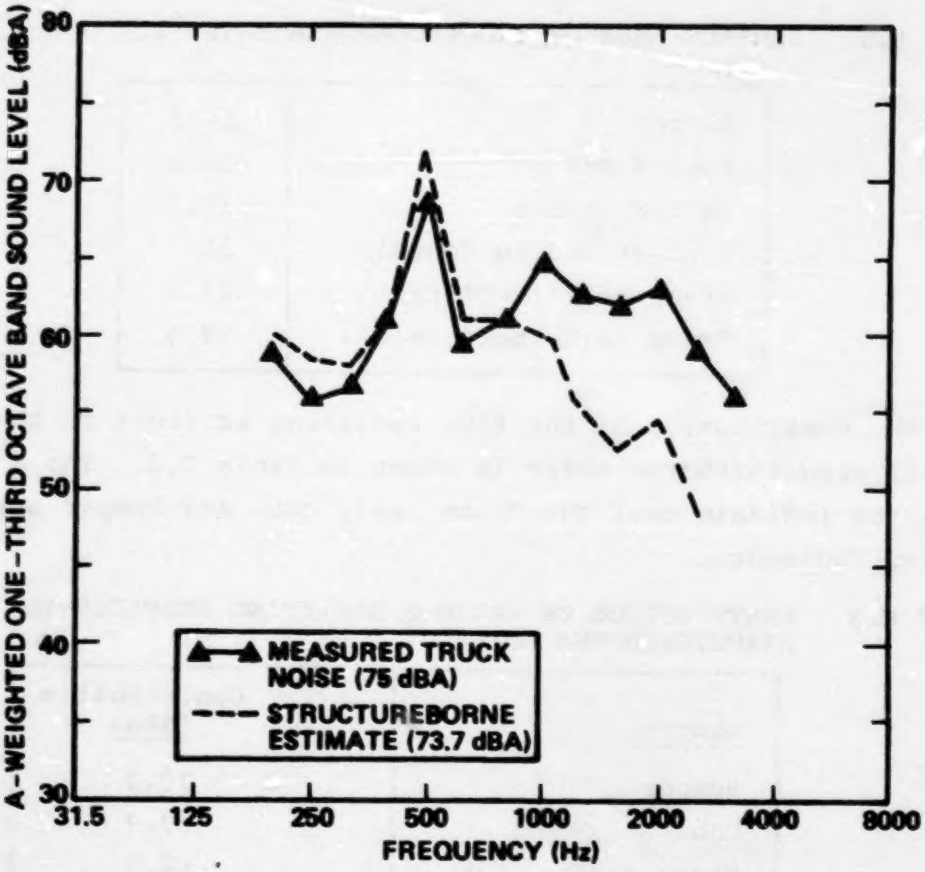

FIG. C.3. COMPARISON OF ESTIMATED STRUCTUREBORNE NOISE AND MEASURED TRUCK NOISE.

include the radiation efficiency of the structural components in the calculation. Consequently, to obtain more accurate estimates of the structureborne noise, we carried out the cover-and-expose measurements described below.

Cover-and-Expose Measurements

To carry out the cover-and-expose measurements, we wrapped the following elements of the truck structure with leaded vinyl and fiberglass:

- Differentials
- Fuel tanks
- Air tank associated with the PTO pump
- Battery boxes
- Cab (excluding the fiberglass hood)
- Frame rails
- Bumper
- Exhaust muffler
- Exhaust pipe outside the enclosure.

We then measured the noise from the truck using the EPA test procedure, first with the truck fully wrapped, and then with the wrapping gradually removed from each of the elements of the structure. For all of these tests the truck was equipped with the original equipment, single-stage mounts, a mockup engine enclosure made of Masonite but similar in geometry to the final aluminum enclosure, and the 15-in. × 10-in. oval muffler used in the final truck configuration. Table C.3 shows the average noise levels from the truck as it was gradually unwrapped.

By taking appropriate logarithmic differences in the noise levels between tests, we can estimate the structureborne sound radiated by each major component of the truck structure. Unfortunately, as Table C.3 shows, the unwrapping of each component resulted in only small increases in noise, on the order of 0.5 dBA. In addition, the noise from the truck was highly variable,

TABLE C.3. MACK R686 NOISE LEVELS WITH VARIOUS COMPONENTS OF THE STRUCTURE WRAPPED.

		Left Side			Right Side		
Configuration No.	Truck Condition	Average Noise Level (dBA)	Std. Dev. (db)	No. of Runs	Average Noise Level (dBA)	Std. Dev. (dB)	No. of Runs
1	Everything wrapped	70.4	0.4	6	70.4	0.2	5
2	Differentials unwrapped	70.3	0.2	5	70.4	0.2	4
3	Fuel tanks and differentials unwrapped	71.8	0.4	6	71.0	0.4	7
4	Fuel tanks, differentials, air tank, and battery boxes unwrapped	72.4	0.6	6	71.7	0.2	7
5	Fuel tanks, differentials, air tank, battery boxes, and cab unwrapped	72.7	0.4	10	72.0	0.2	12
6	Fuel tanks, differentials, air tank, battery boxes, cab, and rear half of frame rail unwrapped	73.2	0.4	5	71.8	0.4	5
7	Fuel tanks, differentials, air tank, battery boxes, cab, and all of frame rail unwrapped	72.9	0.2	2	71.9	0.2	2

as shown by the standard deviations in the table (also on the order of 0.5 dBA). Consequently, the estimation of the structureborne noise from the components of the truck structure by this differencing procedure is subject to some uncertainty. In fact, in the case of the differentials, front portion of the frame rails, and rear portion of the frame rails on the right side of the truck, removal of the wrapping appeared to result in an increase in noise. Similarly, in a later series of tests, removal of the front bumper increased the run by 0.4 dBA. Consequently, a numerical estimate of the noise radiated by those parts of the truck structure is not possible, although one is tempted simply to assume their contributions to be negligible.

Table C.4 shows the estimates of the structureborne sound radiation from the truck, based on the data in Table C.3. Also shown are the preliminary estimates based on the vibration measurements described in the first part of this appendix. For the most part, the preliminary estimates overestimate the sound radiation, a reasonable result, as no attempt was made to account for radiation efficiency or shielding by other elements of the truck, such as the fenders or tires.

A somewhat surprising and not easily explained result in Table C.4 is the fact that the left side of the truck has higher structureborne levels (69.3 dBA), than the right (66.6 dBA). Although the data do show that most of that difference is due to the fuel tanks, it is presently not clear why the left fuel tank should radiate 3 dBA more sound than the right.

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TABLE C.4. STRUCTUREBORNE NOISE FROM STRUCTURAL COMPONENTS OF THE MACK R686.

		Cover-and-Expose Estimates		
Source	Configurations Used from Table C.3	Left Side (dBA)	Right Side (dBA)	Preliminary Estimates (dBA)
Differentials	2 ⊖ 3	-	-	-
Fuel tanks	3 ⊖ 2	66.5	62.1	62.2
Air tank and battery boxes	4 ⊖ 3	63.5	63.4	-
Cab	5 ⊖ 4	60.9	60.2	67.3
Rear portion of frame rail	6 ⊖ 5	63.6	-	66.5
Front portion of frame rail	7 ⊖ 6	-	-	
Bumper	*	-	-	70.2
Total truck structure-borne noise	7 ⊖ 1	69.3	66.6	73.4

*In another series of tests removal of the bumper was found to have a negligible effect on the radiated noise from the truck.